MIL-STD-29A
1 March 1962

SUPERSEDING
MIL-STD-29
23 September 1958

MILITARY STANDARD

SPRINGS, MECHANICAL; DRAWING REQUIREMENTS FOR

APPENDIX

OF REFERENCES TO
MATERIALS, DESIGN FORMULAS,
PROCESSES AND TOLERANCES
FOR MECHANICAL SPRINGS

(APPENDIX INCLUDES NO MANDATORY PROVISIONS)

UNITED STATES
GOVERNMENT PRINTING OFFICE
WASHINGTON: 1962

DEPARTMENT OF DEFENSE

DEFENSE SUPPLY AGENCY

WASHINGTON 25, D. C.

Springs, Mechanical; Drawing Requirements for
MIL-STD-29A

1 March 1962

1. This standard has been approved by the Department of Defense and is mandatory for use by the Departments of the Army, the Navy and the Air Force.

2. In accordance with established procedures, the Ordnance Corps, Bureau of Naval Weapons and the Air Force, have been designated as Army-Navy-Air Force custodians of this standard.

3. Recommended corrections, additions, or deletions should be addressed to the Standardization Division, Defense Supply Agency, Washington 25, D. C.

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MILITARY STANDARD

DRAWING REQUIREMENTS FOR MECHANICAL SPRINGS

1. SCOPE

1.1 SCOPE. This standard illustrates the approved method of drawing, dimensioning, and specifying spring data on detail drawings of each type of mechanical spring described herein. Adherence to this standard is mandatory in the preparation of drawings prepared by or for the Department of Defense. Appendix A contains material, design and manufacturing data representative of modern spring engineering. It is included herein for guidance and convenience only and contains no mandatory provisions.

1.1.1 Types of Springs. The types of springs described in this standard are:

1.1.1.1 Compression.

(a) Helical, Cylindrical; (b) Helical, Stranded Wire; (c) Volute; (d) Coned Disc (Belleville)

1.1.1.2 Extension.

(a) Helical

1.1.1.3 Torsion.

(a) Helical; (b) Torsion bar; (c) Spiral

1.1.1.4 Flat.

(a) Cantilever

1.1.1.5 Constant force.

(a) Flat Strip

1.1.1.6 Garter.

(a) Helical

1.2 PURPOSE. The purpose of this standard is to establish uniform methods of specifying end product data for mechanical springs on drawings prepared by or for the Department of Defense.

2. REFERENCED DOCUMENTS

2.1 NOT APPLICABLE

3. DEFINITIONS

3.1 MECHANICAL SPRING. An elastic body whose mechanical function is to store energy when deflected under load and return the equivalent amount of energy upon being released.

3.2 LOAD. The force exerted upon or by a spring in order to reproduce or modify motion or to maintain a force system in equilibrium. A static load is slowly applied and does not subject a spring to fatigue. A dynamic load is either a suddenly applied load or one that causes impact due to kinetic energy. It also may be frequently repeated, thus shortening fatigue life.

3.3 TOTAL DEFLECTION. The movement of a spring from free position to its maximum operating position. In a compression spring, it is the deflection from the Free Length to the Solid Length.

3.4 DEFLECTION PER COIL. The total deflection of the spring divided by the number of active coils.

3.5 SET. The permanent distortion from the manufactured dimensions which occurs when the spring is stressed beyond the elastic limit of the material.

3.6 ACTIVE COILS. The number of coils which are used in computing the total deflection of the spring.

3.7 TOTAL COILS. The number of active

coils plus the coils forming the ends, (Compression Springs).

3.8 SOLID LENGTH. The overall length of a compression spring when all coils are fully compressed.

3.9 FREE LENGTH. The overall length of a spring in the unloaded position.

3.10 PITCH. The distance between centers of adjacent active coils of a spring in the unloaded position.

3.11 INITIAL TENSION. A load wound into certain helical extension springs during the coiling operation which keeps the coils tightly closed and which must be exceeded by an applied load before the coils begin to open.

3.12 SPRING INDEX. The ratio of the mean spring diameter to the diameter of the wire.

$$\text{Spring index} = D/d = \frac{OD - d}{d} = \frac{ID + d}{d}$$

3.13 SPRING RATE. The load required to deflect a compression or extension spring one inch or the load required to deflect a torsion spring one degree or one revolution. Also referred to as "Scale," "Gradient" and "Load Factor."

3.14 TORQUE. A turning force about an axis multiplied by the distance from the load to the axis, usually expressed in pound-inches (LB IN.) or in ounce-inches (OZ IN.) and always used in conjuncion with the number of degrees of rotation, number of revolutions or deflected position.

3.15 SPRING TOLERANCE. The permissible variations from a given dimension.

3.16 FACTOR OF SAFETY. The ratio of the maximum load a spring can sustain without permanent set to the maximum applied load.

3.17 DIRECTION OF HELIX. When a spring is viewed from one end, the direction of helix is right hand when the coil recedes in a clockwise direction, and left hand when it recedes in a counterclockwise direction. A right hand helix follows the same direction as the threads on a standard screw. A left hand helix is most popular for compression and extension springs.

3.18 REFERENCE DIMENSION. A reference dimension is a dimension without tolerance which is entered on a drawing for informational purposes and which does not directly govern manufacturing or inspection operations. Reference dimensions are indicated on drawings by writing the abbreviation REF directly following or under the dimension.

3.19 STRESS RELIEVE. To subject springs to a low temperature heat treatment after coiling or bending to remove residual stresses caused by the forming operation. Also called "strain relieve", "stress equalizing", "tempering", "blueing" and "baking".

3.20 RANGE OF STRESS. The difference between the stress at maximum and minimum loads.

3.21 HARDEN. To cause a material to become harder by heat treatment. Spring steels are heated above the critical temperature and quenched in oil. Precipitation hardening alloys such as Beryllium-Copper, K-Monel and Inconel X are heated at elevated temperatures for extended periods of time.

3.22 ENDURANCE LIMIT. The maximum stress or range of stress at which a material will operate indefinitely without failure. This limit for springs is usually determined at 10,000,000 cycles of deflection.

3.23 ELASTIC LIMIT. The maximum stress to which a material may be subjected without causing a permanent set. Also the stress beyond which a material must be subjected to form a coil, radius or bend.

3.24 **MODULUS OF RIGIDITY.** The measure of elastic ability. A mathematical constant expressing the stiffness, rigidity, elasticity and flexibility of a material. Also called "modulus of elasticity" and "Young modulus".

4. GENERAL REQUIREMENTS

4.1 Not Applicable

5. DETAIL REQUIREMENTS

5.1 **GENERAL.** Figures 2(a) through 2(d), 4(a) through 4(d), 7, 10, 12, 13, 14, 16, 17 and 18 are Drawing Requirement Charts illustrating approved methods of drawing and dimensioning the various types of springs listed in paragraph 1.1.1 and include compilations of spring data to be used as applicable when prescribing manufacturing information on detail drawings of mechanical springs. The inclusion herein of these charts should not be interpreted as restricting the scope of spring characteristics to be specified on engineering drawings delineating springs. The specification of correct and complete manufacturing data is the responsibility of the cognizant Design Activity. Figures 19 thru 30 are examples of typical detail drawings of springs illustrating the application of the information included on the charts.

5.1.1 **Delineation.** The simplified drafting methods shown on the Drawing Requirement Charts and the typical detail drawings, e.g., Figures 2(a) and 19, are generally satisfactory for depicting helical springs.

5.2 SPECIFICATIONS APPLICABLE TO ALL SPRINGS:

5.2.1 **Material specifications.** Material Specifications are designated in the space provided in the title block on the drawing. When such space is inadequate enter therein "SEE NOTE" and describe the material requirements in a general note.

5.2.2 **Diameter of wire.** Wire diameter, or width an thickness of material, shall be designated in decimals, or common fractions of an inch, as applicable, and unless otherwise specified, all dimensions apply before plating or applying finishes. Such dimensions are used to avoid confusion resulting from various gage numbering systems in current use. However, they should conform with the appropriate gage customarily used with the material specified, whenever possible. For oil tempered and corrosion resisting steels such diameters should conform with the U. S. Steel Wire Gage, which is the same as the Washburn & Moen Gage. For non-ferrous materials the American Wire Gage, which is the same as the Brown & Sharpe Gage, is used. Music Wire is drawn to the American Steel Wire and Music Wire Gage but other gages and intermediate sizes are available. Do not specify the name of the Gage used. Tolerances on diameter and thickness shall not be specified when they are covered in the appropriate material specifications or by general tolerance notes. Specify a tolerance on the width, as applicable. See Table I, Preferred Sizes of Spring Materials, in Appendix A, Section I.

5.2.3 **Inspection Notes.** Detail drawings of springs that are critical, including those springs subjected to critical conditions of temperature, stress or corrosive environments, shall include appropriate instruction notes applicable to controlling the manufacture within the desired environments. It is appropriate to specify those spring factors requiring inspection, the tests required, and whether all springs produced from the particular drawing shall be inspected. Consistent with Quality Control Procedure the size of inspection samples shall not be specified.

5.3 FORMAT FOR SPECIAL DATA NOTES.

When notes prescribing special data are required on detail drawings of springs, it is recommended that the following format be used:

(a) Stress at ....°F ± ....°F for .... minutes, after coiling (or forming).

(b) Hardness range Rc .... to ....

(c) Squareness of ends in free position within ....°.

(d) Cold set to solid.

(e) Shot peen (give specification) to .... intensity.

(f) Protective coating (give specification).

(g) Non destructive inspection (give specification).

(h) The body shall not camber more than .... IN. in its entire length.

(i) Test over arbor dia .... IN.

(j) To withstand temperatures of .... °F for .... hours with loss of load not to exceed ....%.

(k) To withstand .... deflections from initial to final position with loss of load not to exceed..........%.

5.4 HELICAL COMPRESSION SPRINGS

5.4.1 Definition. A compression spring is an open-coil helical spring that offers resistance to a compressive force applied axially. Compression springs are generally made cylindrical in form, although other forms are used, such as conical, tapered, concave or convex, consistent with design requirements. Because of ready supply, wire of circular cross section is used whenever possible; however, where advantageous to design, wire of square or rectangular cross section may be utilized. Figure 1 illustrates several forms of helical compression springs.

5.4.2 Drawing Requirements for Helical Compression Springs. Guide lines for specifying pertinent dimensional and load data on engineering drawings of helical compression springs, which satisfy specific design requirements and, simultaneously, allow maximum latitude in manufacturing, are provided herein, categorized as follows:

(1) *No Load Specified.* The design activity assumes responsibility for the load capacity of the spring. The manufacturer is required to furnish a spring meeting the dimensional data specified. The free length, the coil diameter, and the total number of coils are specified, each with a tolerance. The tolerance on the wire diameter conforms to the governing material specification. Figure 2(a) shows suggested data applicable to springs in the "no load" category.

(2) *One Load Specified.* This category has application when the spring is required to develop a load, within a specified tolerance, preferably at the initial assembled length. The spring, normally, is not subjected to further deflection in operation. The manufacturer is required to meet the load requirement, but the free length and the total number of coils are not restricted, being designated REF. The tolerance on the wire diameter conforms to the governing material specification and the manner of coil diameter callout is dependent upon spring function. Figure 2(b) shows suggested data applicable to springs in the "one load" category.

(3) *Two Loads Specified.* This category has application when the spring is required to develop a load, within a specified tolerance, at each of two definite compressed lengths, normally at the initial and the final operating positions in the assembly. The manufacturer is required to meet the load requirements, other spring characteristics being designated as described in paragraph 5.4.2(2). Figure 2(c) shows suggested data applicable to springs in the "two loads" category.

(4) *Spring Rate Specified.* This category

has application in assemblies in which the spring rate is the most significant characteristic, for example, in calibrated scales. The manufacturer is required to meet a prescribed spring rate, within a specified tolerance, but the free length and the total number of coils are permitted to vary, being designated REF. The tolerance on the wire diameter conforms to the governing material specification and the manner of coil diameter callout is dependent upon the spring function. Additionally, when closer control of the load to be developed at the initial assembled length is desired, one load, with a tolerance, to be developed at the assembled length should be specified. Figure 2(d) shows suggested data applicable to springs in the "spring rate" category.

5.4.2.1 Coil Diameter. Depending upon the application of the spring, specify one of the following:

(a) TO WORK OVER....... IN. (MAX) DIA ROD

(b) TO WORK IN IN. (MIN) DIA BORE

(c) ID, with tolerance

(d) OD, with tolerance

5.4.2.2 Direction of Helix. When governed by design requirements, specify the direction of helix as "LEFT HAND" (or LH), or "RIGHT HAND" (or RH), as applicable; otherwise specify as OPTIONAL. In most cases the hand is not important except when a plug screws into an end; or when one spring fits inside another, in which case one spring should be designated left hand and the other right hand.

5.4.2.3 Type of Ends. The types of ends having application to helical compression springs are illustrated in Figure 3. Specify the type of ends on the drawing; when necessary, the ends should be dimensioned.

5.4.2.4 Solid Length. The maximum solid length shall be specified for any spring category when this parameter is essential to design requirements. However, solid length callout should be omitted whenever practicable. Except when necessary to satisfy function, springs should not be designed to go solid in operation.

HELICAL COMPRESSION SPRING FORMS

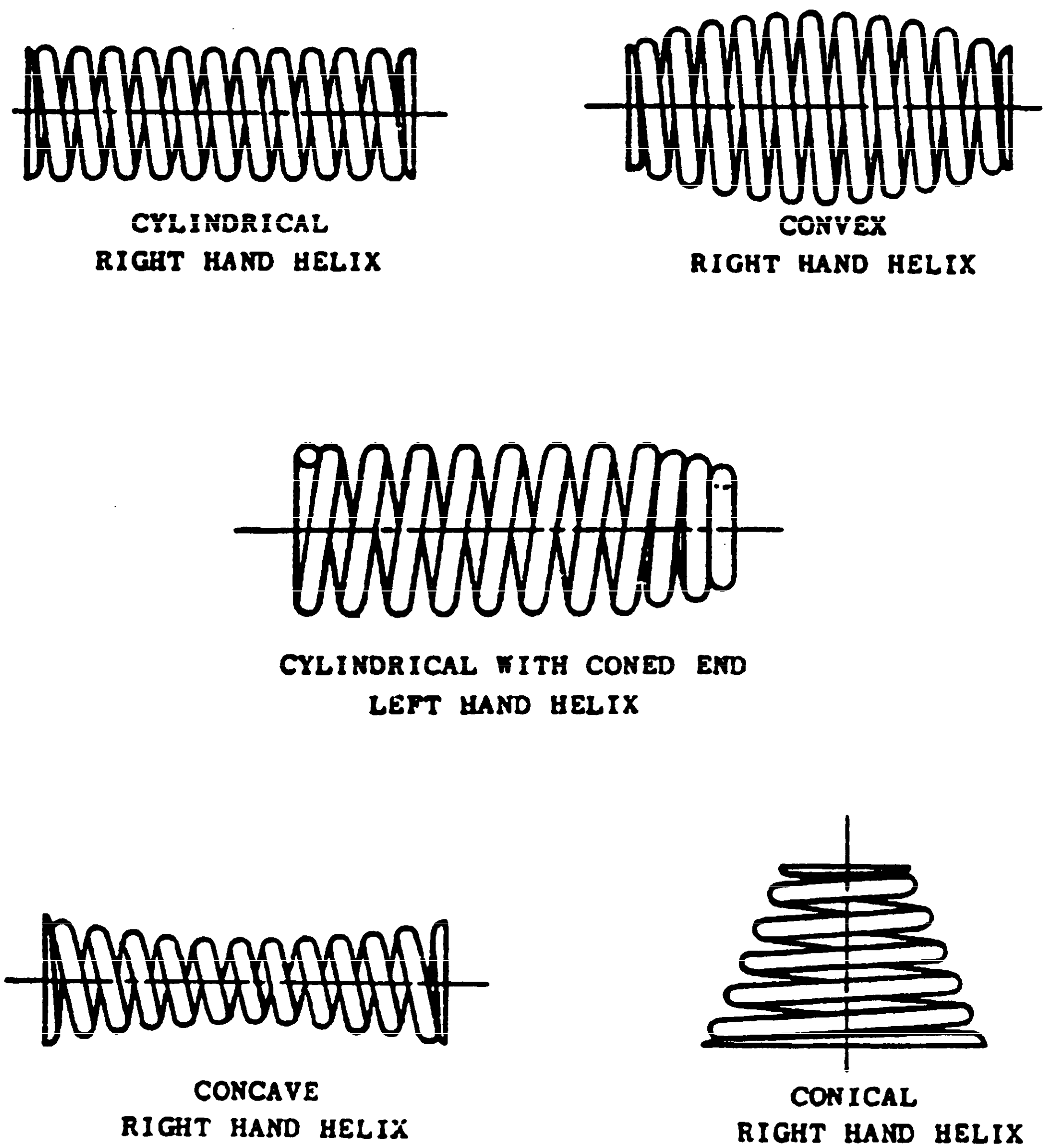
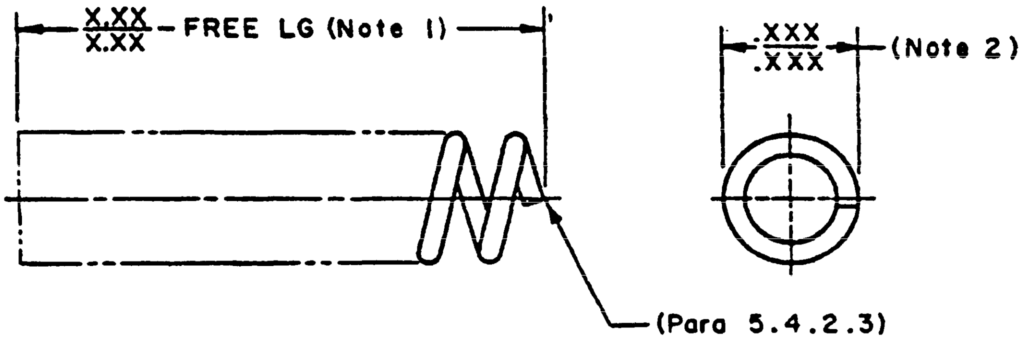


FIGURE 1

DRAWING REQUIREMENTS FOR HELICAL COMPRESSION SPRINGS

WHEN NO LOAD IS SPECIFIED



Spring data

		Para
MATERIAL SPECIFICATION		5.2.1
WIRE DIA	XXX IN.	5.2.2
DIRECTION OF HELIX		5.4.2.2
TOTAL COILS XX ± XX		Note 3
SPECIAL DATA		5.3

Note 1: Specify free length, with tolerance (or by limits). (Refer to Appendix A, Section III, Table XIX.)

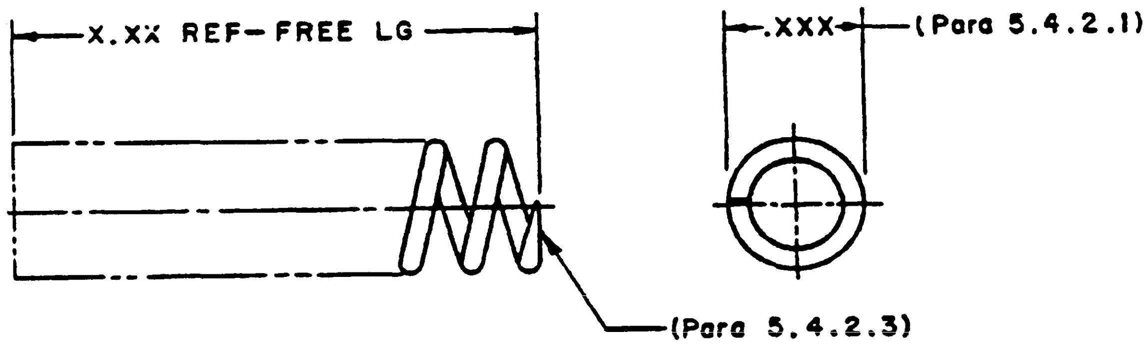
Note 2: When spring operates over guide, specify inside diameter of coil, with tolerance (or by limits). For other applications, specify outside diameter with tolerance (or by limits). (Refer to Appendix A, Section III, Table XVI.)

Note 3: Specify total number of coils, with tolerance. (Refer to Appendix A, Section III, Table XVII.)

FIGURE 2(a)

DRAWING REQUIREMENTS FOR HELICAL COMPRESSION SPRINGS

WHEN ONE LOAD IS SPECIFIED



Spring data

		Para
MATERIAL SPECIFICATION		5.2.1
WIRE DIA	.XXX IN.	5.2.2
DIRECTION OF HELIX		5.4.2.2
TOTAL COILS	XX REF	Note 1
LOAD AT COMPRESSED LG OF XXX IN. — XX IB ± XX LB		Note 2
SPECIAL DATA		5.3

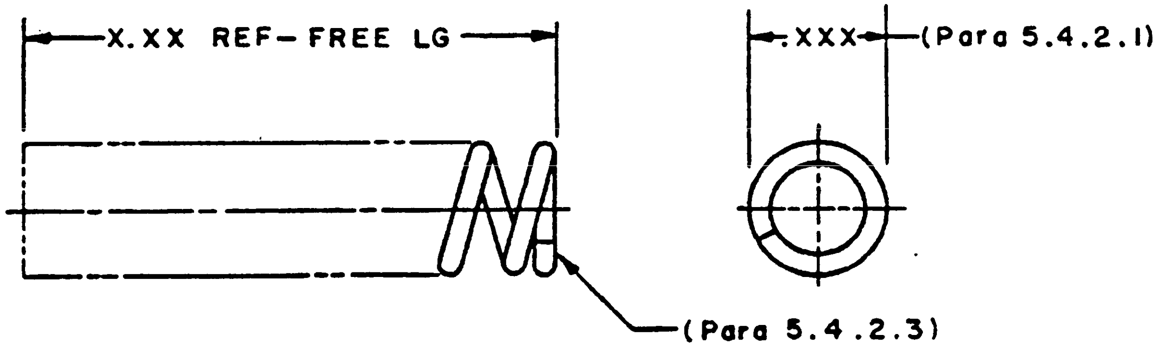
Note 1: Specify total number of coils; designate REF.

Note 2: Specify the *one load*, with tolerance, to be developed at a definite length, preferably the initial assembled length.

FIGURE 2(b)

DRAWING REQUIREMENTS FOR HELICAL COMPRESSION SPRINGS

WHEN TWO LOADS ARE SPECIFIED



Spring data

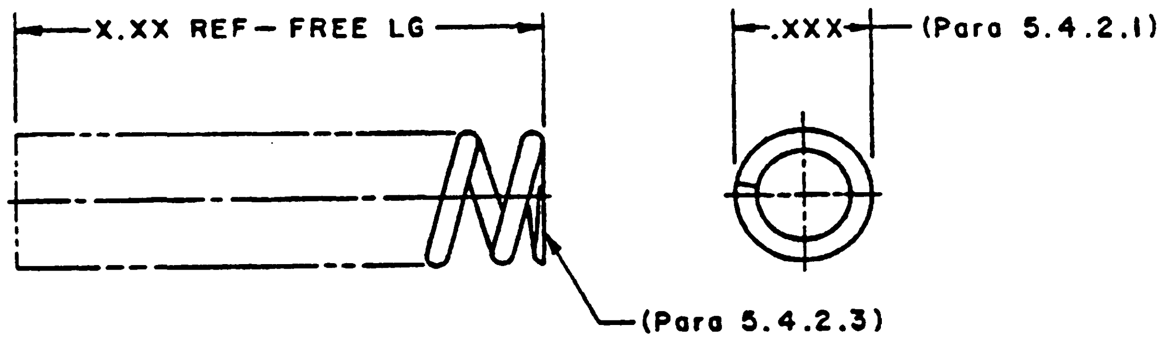
		Para
MATERIAL SPECIFICATION		5.2.1
WIRE DIA	.XXX IN.	5.2.2
DIRECTION OF HELIX		5.4.2.2
TOTAL COILS	XX REF	Note 1
LOAD AT COMPRESSED LG OF X.XX IN – XX LB ± XX LB		Note 2
LOAD AT COMPRESSED LG OF X.XX IN – XX LB ± XX LB		Note 2
SPECIAL DATA		5.3

Note 1: Specify total number of coils; designate REF.

Note 2: Specify load to be developed at each of two definite compressed lengths, preferably at the initial and the final operating positions in the assembly.

FIGURE 2(c)

DRAWING REQUIREMENTS FOR HELICAL COMPRESSION SPRINGS WHEN SPRING RATE IS SPECIFIED



Spring data

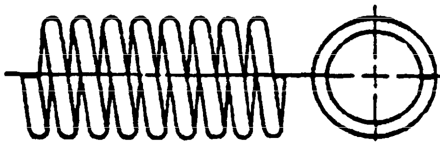
		Para
MATERIAL SPECIFICATION		5.2.1
WIRE DIA	.XXX IN.	5.2.2
DIRECTION OF HELIX		5.4.2.2
TOTAL COILS	XX REF	Note 1
LOAD AT COMPRESSED LG OF X.XX IN. — XX LB ± XX LB		5.4.2(4)
SPRING RATE XX LB/IN. ± XX LB/IN. BETWEEN COMPRESSED LG OF XX IN. AND XX IN.		Note 2
SOLID LG WITHOUT PERMANENT SET	XX IN.	5.4.2.4
SPECIAL DATA		5.3

Note 1: Specify total number of coils; designate REF.

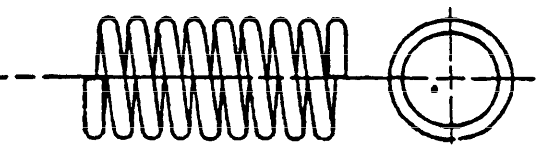
Note 2: Specify spring rate, with tolerance.

FIGURE 2(d)

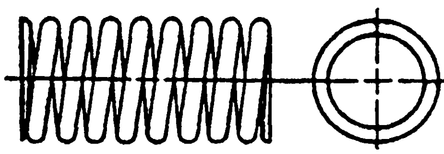
TYPES OF HELICAL COMPRESSION SPRING ENDS



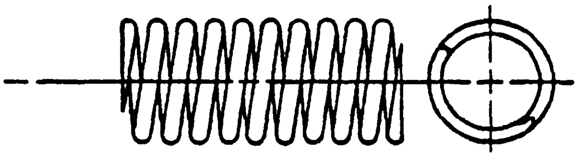
OPEN ENDS NOT GROUND
RIGHT HAND HELIX



CLOSED ENDS NOT GROUND
RIGHT HAND HELIX



CLOSED ENDS GROUND
LEFT HAND HELIX



OPEN ENDS GROUND
LEFT HAND HELIX

FIGURE 3

5.5 STRANDED WIRE COMPRESSION SPRINGS

5.5.1 Definition. Stranded wire compression springs are helical compression springs formed from two, three or more wires twisted about each other, or about one wire as a core, to form a single strand. Stranded wire compression springs have the advantage of damping the migratory waves that traverse the spring under shock loading. Such springs should be specified with caution since stranded wire is specially made and is not readily available.

5.5.2 Drawing Requirements for Stranded Wire Compression Springs. Figure 20 is a typical detail drawing of a stranded wire helical compression spring. The methods of drawing and dimensioning, and the spring data applicable to stranded wire springs correspond in general to those shown in Figures 2(c) for compression springs, except that the additional spring information noted herein is required on detail drawings:

(a) Number of Wires in Strand.

(b) Diameter of each Wire.

(c) Length of Lay (Distance parallel to strand axis in which a single wire makes one turn.)

(d) Direction of Strand Helix
NOTE: The strand helix shall be opposite in direction to the coil helix.

(e) Direction of Coil Helix (Left Hand or Right Hand).

(f) **Type of Ends.** Specify either CLOSED ENDS NOT GROUND or OPEN ENDS NOT GROUND, as applicable. Specify that ends are to be soldered, brazed or welded to prevent unraveling, and sharp burrs shall be removed.

5.6 HELICAL EXTENSION SPRINGS

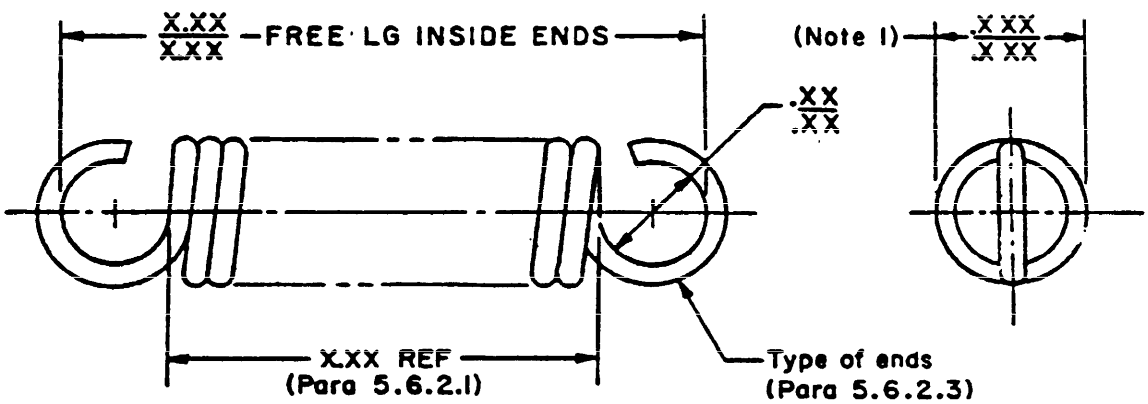
5.6.1 Definition. A helical extension spring is a close-wound spring with or without initial tension, or an open-wound spring that offers resistance to an axial force tend-ing to extend its length. Extension springs are formed or fitted with ends which are used for attaching the spring to the assembly. They are generally made of wire of circular cross-section, but when advantageous to design, may be made of wire of square or rectangular cross-section. Extension springs may be wound with adjacent coils touching so lightly that deflection begins the instant a load is applied. However, they are usually designed to retain about 10% of their maximum applied load as initial tension. This holds the coils together thus making it easier to coil the springs and meet test requirements. They may also be wound so tightly, that a large portion of the applied load (up to about 33% of the maximum load) must be expended before actual deflection occurs. Figure 5 illustrates various forms of helical extension springs.

5.6.2 **Drawing Requirements for Helical Extension Springs.** Guidelines for specifying dimensional and load data on engineering drawings showing helical extension springs are similar to those established for helical compression springs. Figures 4(a) through 4(d) show suggested guidelines for use, selectively, in delineating helical extension springs, in the categories described in paragraph 5.4.2.

5.6.2.1 **Total Coils and Length ove Coils.** All coils in an extension spring usually are active. Exceptions are those with plug ends and those with end coils coned over swivel hooks. Specify the total number of coils and the length over coils each as REF, except as noted in Figure 4(a).

DRAWING REQUIREMENTS FOR HELICAL EXTENSION SPRINGS

WHEN NO LOAD IS SPECIFIED



Spring data

		Para
MATERIAL SPECIFICATION		5.2.1
WIRE DIA	X.XX IN.	5.2.2
DIRECTION OF HELIX		5.4.2.2
TOTAL COILS XX ± XX		Note 2
RELATIVE POSITION OF ENDS		5.6.2.4
EXTENDED LG INSIDE ENDS WITHOUT PERMANENT SET X.XX IN. (MAX.)		5.6.2.2
INITIAL TENSION XX LB ± XX LB		Note 3
SPECIAL DATA		5.3

Note 1: Specify outside diameter of coil, with tolerance (or by limits).
(Refer to Appendix A, Section III, Table XVI)

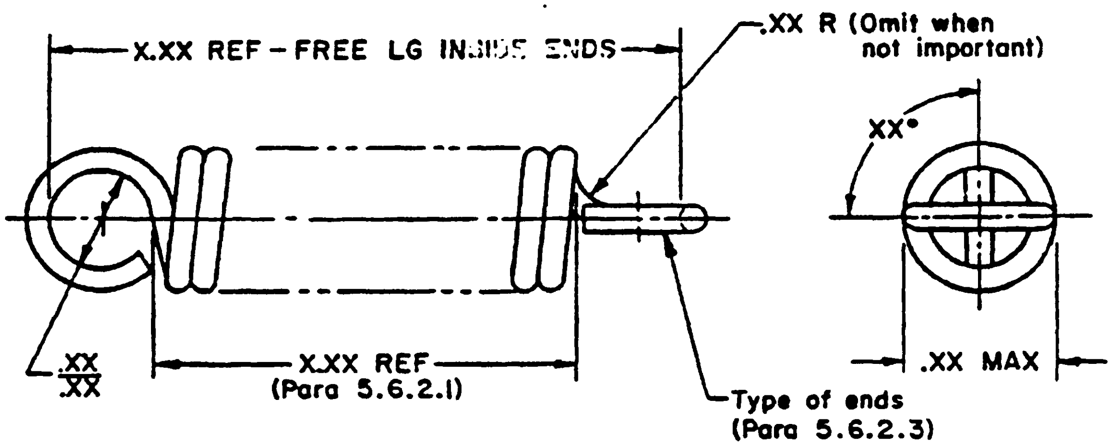
Note 2: Specify total number of coils with tolerance.
(Refer to Appendix A, Section III, Table XVII)

Note 3: When essential to design requirements, specify initial tension with tolerance; otherwise specify "INITIAL TENSION OPTIONAL."

FIGURE 4(a)

DRAWING REQUIREMENTS FOR HELICAL EXTENSION SPRINGS

WHEN ONE LOAD IS SPECIFIED



Spring data

		Para
MATERIAL SPECIFICATION		5.2.1
WIRE DIA	.XXX IN.	5.2.2
DIRECTION OF HELIX		5.4.2.2
TOTAL COILS	XX REF	Note 1
RELATIVE POSITION OF ENDS		5.6.2.4
EXTENDED LG INSIDE ENDS WITHOUT PERMANENT SET X.XX IN. (MAX.)		5.6.2.2
INITIAL TENSION XX LB ± XX LB		Note 2
LOAD XX LB ± XX LB AT X.XX IN. EXTENDED LG INSIDE ENDS		Note 3
SPECIAL DATA		5.3

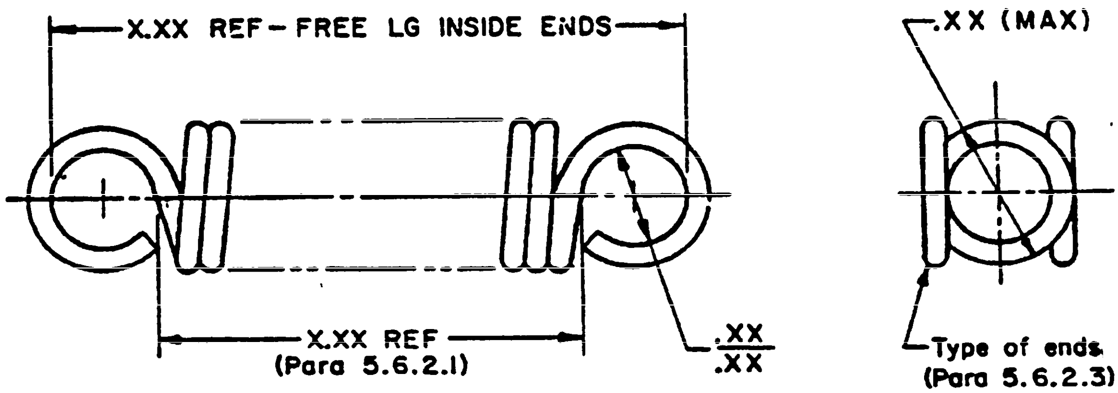
Note 1: Specify total number of coils; designate REF.

Note 2: When essential to design requirements, specify initial tension with tolerance; otherwise specify INITIAL TENSION OPTIONAL."

Note 3: Specify the *one load*, with tolerance, to be developed at a definite extended length, preferably the initial assembled length.

FIGURE 4(b)

DRAWING REQUIREMENTS FOR HELICAL EXTENSION SPRINGS WHEN TWO LOADS ARE SPECIFIED



Spring data

	Para
MATERIAL SPECIFICATION	5.2.1
WIRE DIA .... XXX IN.	5.2.2
DIRECTION OF HELIX	5.4.2.2
TOTAL COILS .... XX REF	Note 1
RELATIVE POSITION OF ENDS	5.6.2.4
EXTENDED LG INSIDE ENDS WITHOUT PERMANENT SET X.XX IN. (MAX.)	5.6.2.2
INITIAL TENSION X LB ± X LB	Note 2
LOAD XX LB ± XX LB AT XX IN. EXTENDED LG INSIDE ENDS	Note 3
LOAD XX LB ± XX LB AT XX IN. OPERATING LG INSIDE ENDS	Note 3
SPECIAL DATA	5.3

Note 1: Specify total number of coils; designate REF.

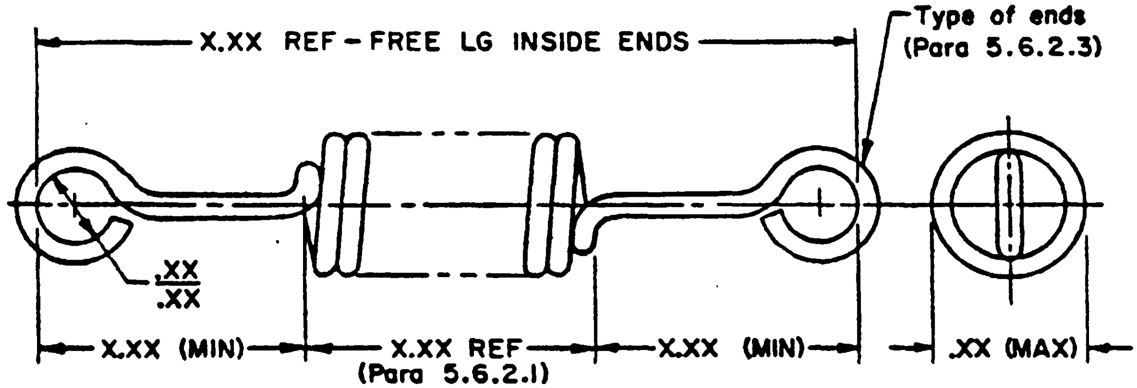
Note 2: When essential to design requirements, specify initial tension with tolerance; otherwise specify "INITIAL TENSION OPTIONAL."

Note 3: Specify the load to be developed at each of two definite extended lengths, preferably at the initial and the final operating lengths in the assembly.

FIGURE 4(c)

DRAWING REQUIREMENTS FOR HELICAL EXTENSION SPRINGS

WHEN SPRING RATE IS SPECIFIED



Spring data

		Para
MATERIAL SPECIFICATION		5.2.1
WIRE DIA	.XXX IN.	5.2.2
DIRECTION OF HELIX		5.4.2.2
TOTAL COILS	XX REF	Note 1
RELATIVE POSITION OF ENDS		5.6.2.4
EXTENDED LG INSIDE ENDS WITHOUT PERMANENT SET XX IN. (MAX.)		5.6.2.2
INITIAL TENSION XX LB ± XX LB		Note 2
SPRING RATE XX LB/IN. ± XX LB/IN. BETWEEN EXTENDED LG OF X.XX IN. AND X.XX IN. INSIDE ENDS		Note 3
LOAD XX LB ± XX LB AT XX IN. OPERATING LG INSIDE ENDS		5.4.2(4)
SPECIAL DATA		5.8

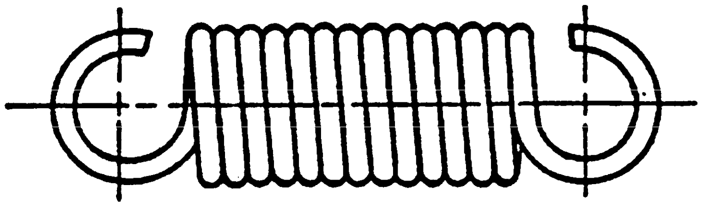
Note 1: Specify total number of coils; designate REF.

Note 2: When essential to design requirements, specify initial tension with tolerance; otherwise specify "INITIAL TENSION OPTIONAL."

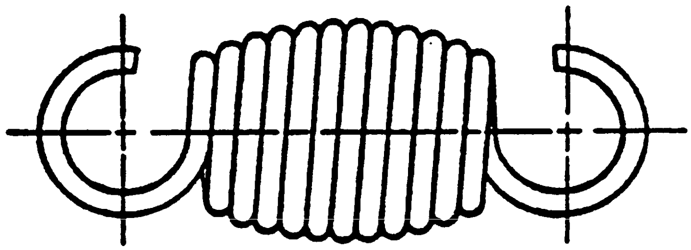
Note 3: Specify spring rate with tolerance.

FIGURE 4(d)

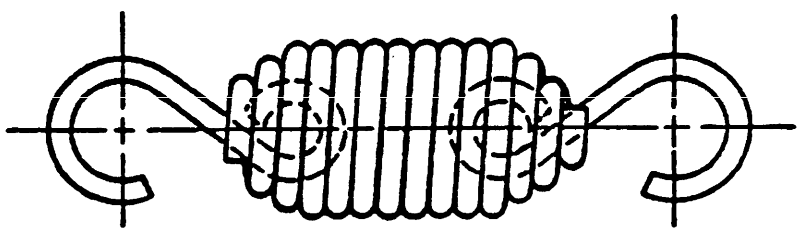
HELICAL EXTENSION SPRING FORMS



CYLINDRICAL SHAPE
RIGHT HAND HELIX



CONVEX SHAPE
LEFT HAND HELIX



CONED ENDS WITH SHORT SWIVEL HOOKS
LEFT HAND HELIX

FIGURE 5

TYPES OF HELICAL EXTENSION SPRING ENDS

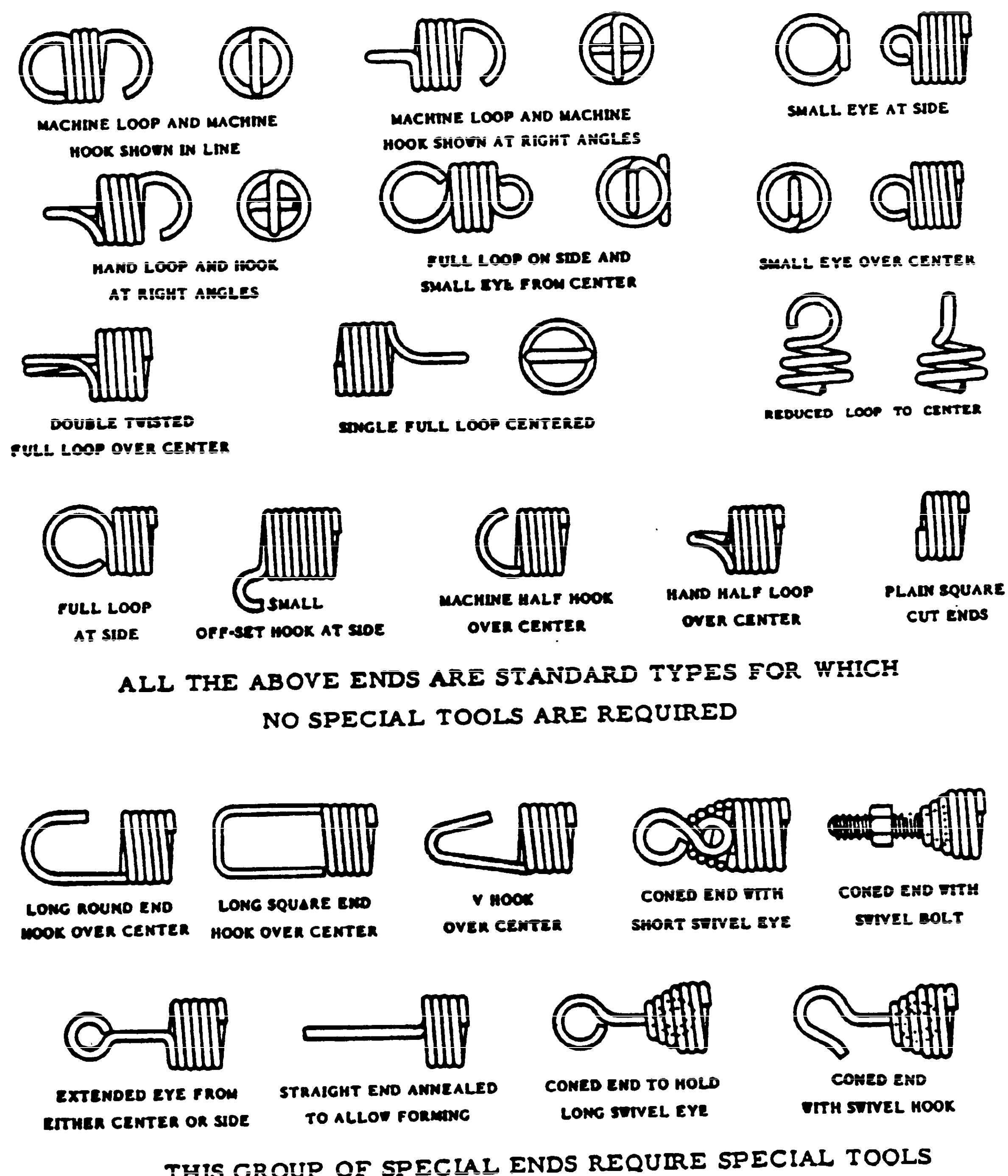


FIGURE 6

5.6.2.2 Maximum Extended Length. Specify the maximum allowable extended length without permanent set as a protection against over extending the spring during assembly only if essential to design requirements. Be sure the total stress at this deflection is less than the elastic limit in torsion.

5.6.2.3 Types of Ends. Typical types of ends applicable to helical extension springs are illustrated in Figure 6. The types of ends required shall be completely delineated and dimensioned on the drawing. Usually the outside diameter of a hook or loop is the same as the outside diameter of the spring. A "Hook" has an open space between its end and the body of the spring. A "Loop" is a closed hook.

5.6.2.4 Relative Position of Ends. When controlled by assembly requirements, the relative position of the ends shall be specified with a tolerance; otherwise include in the spring data this note:

RELATIVE POSITION OF ENDS UNIMPORTANT.

5.7 HELICAL TORSION SPRINGS

5.7.1 Definition. Helical torsion springs are springs that offer resistance to or exert a turning force in a plane at right angles to the axis of the coil. Torsion springs are generally made of circular cross section in a variety of forms and are employed in applications seldom alike. Figures 8 and 9 illustrate various forms of torsion springs and some of the ends used on such springs.

5.7.2 Drawing Requirements for Helical Torsion Springs. Explanation of the spring data listed in Figure 7 is given below. Figure 22 is a typical drawing of a torsion spring.

5.7.2.1 Total Coils and Length over Coils. All coils in a torsion spring usually are active. Specify the total number of coils and the length over coils in the free position each as REF, or with tolerances, if essential to the design.

5.7.2.2 Direction of Helix. The helix of a torsion spring is important; specify as LEFT HAND or RIGHT HAND, as applicable Helical torsion springs should be so designed that the applied loads tend to wind up the spring.

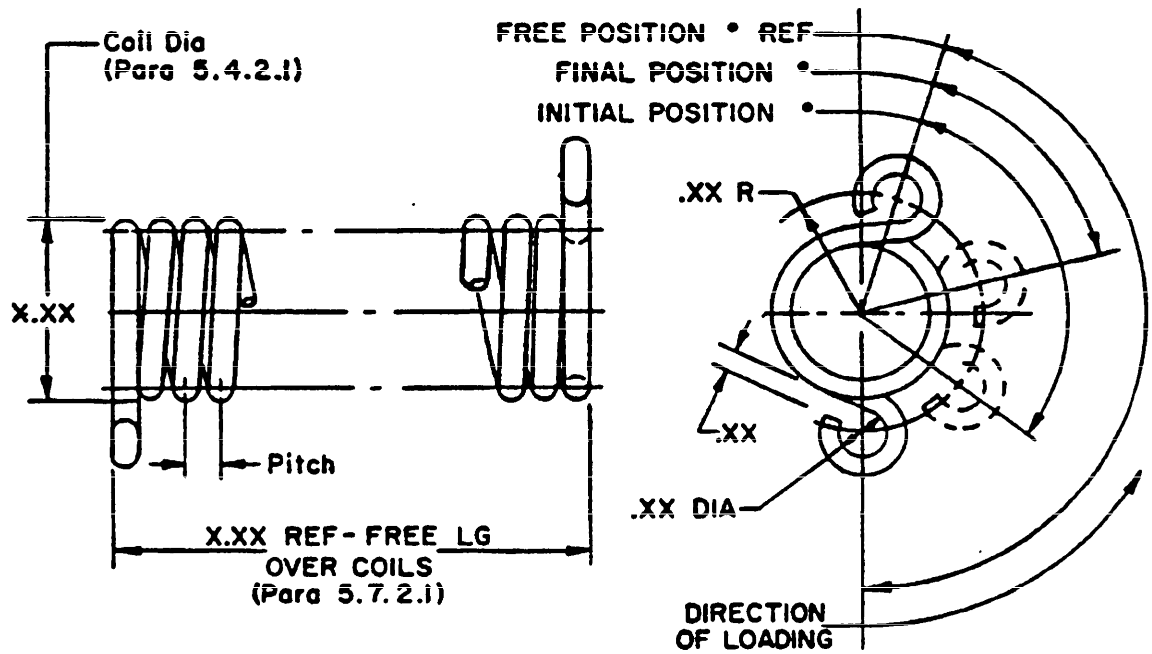
5.7.2.3 Pitch of Coils or Initial Tension. When open coils are used specify the pitch as a REFERENCE dimension; except that when initial tension is required, specify the initial tension, in pounds with a tolerance, instead of pitch. Initial tension is seldom desirable in torsion springs.

5.7.2.4 Torque Loads. Specify the torque loads at the initial and final positions between the deflected end and not at deflection from free position. However, if more than one revolution is required, specify the number of revolutions from the free position. Example: 40 lb in. ± 4 lb in. at 2.5 revolutions from free position. Tolerances shall be applied to the torque loads but not to the angle between the ends. Also state under SPECIAL DATA the diameter of the arbor or rod over which the spring is tested, which should be the same size as the shaft over which it is operated.

5.7.2.5 Maximum Deflection without Permanent Set. Specify the maximum allowable deflection without permanent set in degrees of rotation beyond the final position, as a precaution against overstressing the spring during assembly only if essential to design requirements. Be sure the total stress at this deflection is less than the elastic limit in bending.

5.7.2.6 Spring Rate. Specify the spring rate in pound inches per degree of deflection as a REFERENCE except where particular design requirements necessitate a tolerance. Examples of such applications are in cali-

DRAWING REQUIREMENTS FOR HELICAL TORSION SPRINGS



Spring data

		Para
MATERIAL SPECIFICATION		5.2.1
WIRE DIA	XXX IN.	5.2.2
DIRECTION OF HELIX		5.7.2.2
TOTAL COILS	XX REF	5.7.2.1
PITCH	XX IN. REF	5.7.2.3
TORQUE XX LB IN. ± XX LB IN. AT INITIAL POSITION		5.7.2.4
TORQUE XX LB IN. ± XX LB IN. AT FINAL POSITION		5.7.2.4
MAXIMUM DEFLECTION WITHOUT SET BEYOND FINAL POSITION	XX DEG	5.7.2.5
SPRING RATE	XX LB IN./DEG REF	5.7.2.6
TYPES OF ENDS		5.7.2.7
SPECIAL DATA		5.8

FIGURE 7

TYPES OF HELICAL TORSION SPRING ENDS

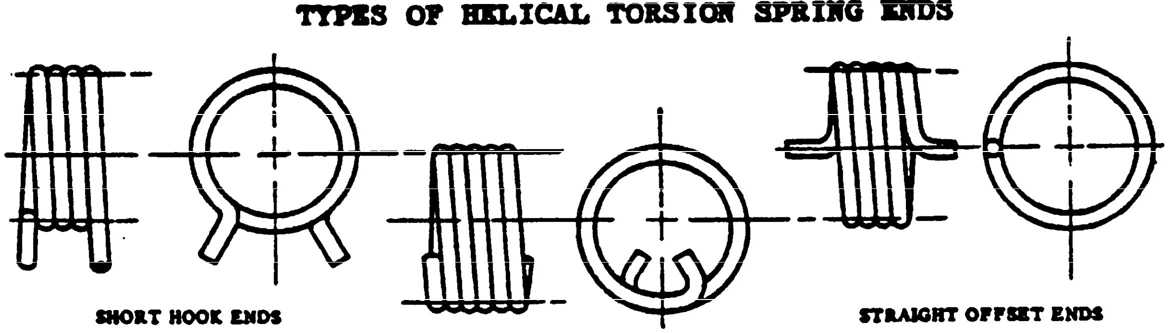


FIGURE 8

HELICAL TORSION SPRING FORMS

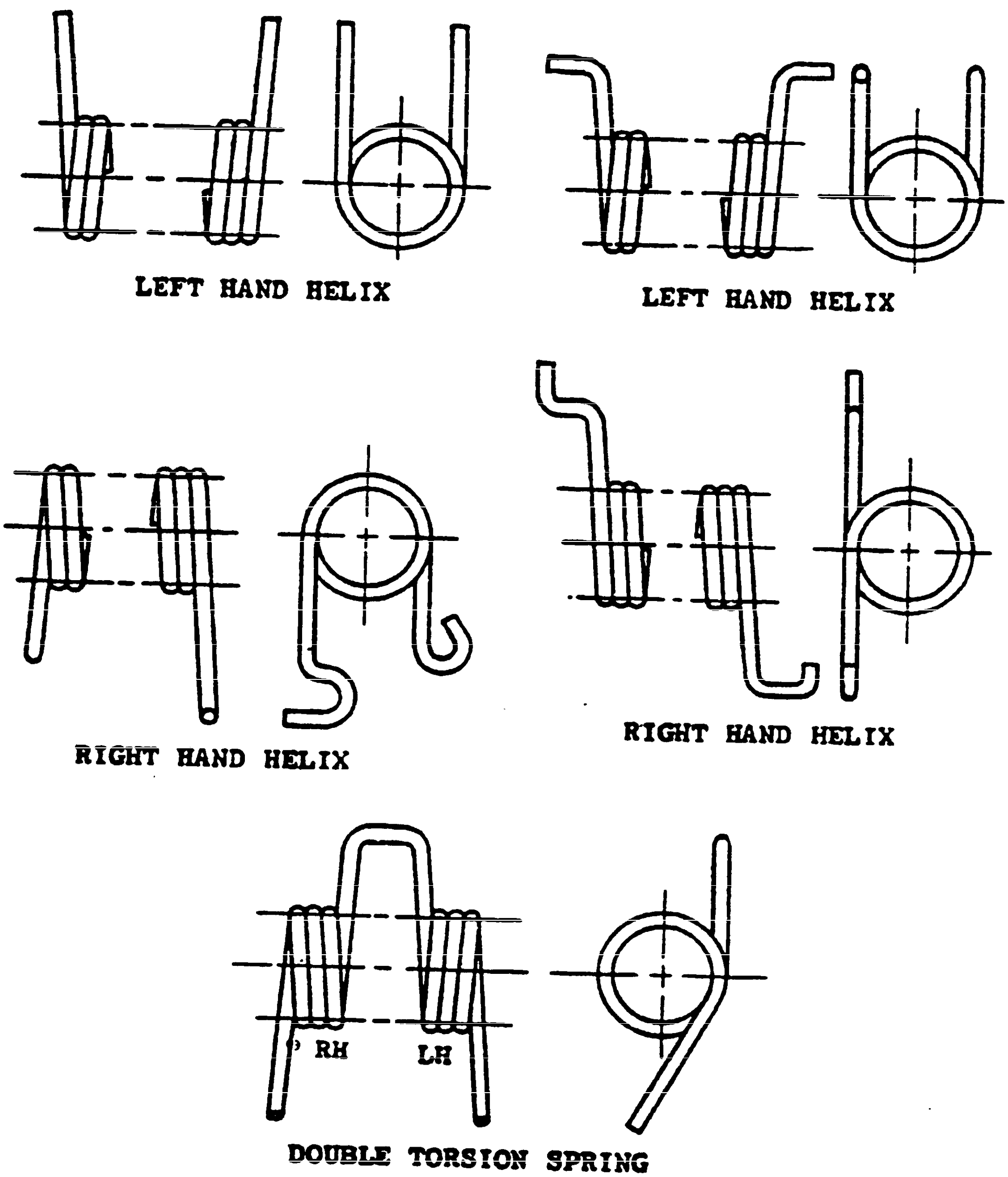


FIGURE 9

brated scales or instruments. When the spring rate is specified with a tolerance designate the loads at deflected positions as a REFERENCE.

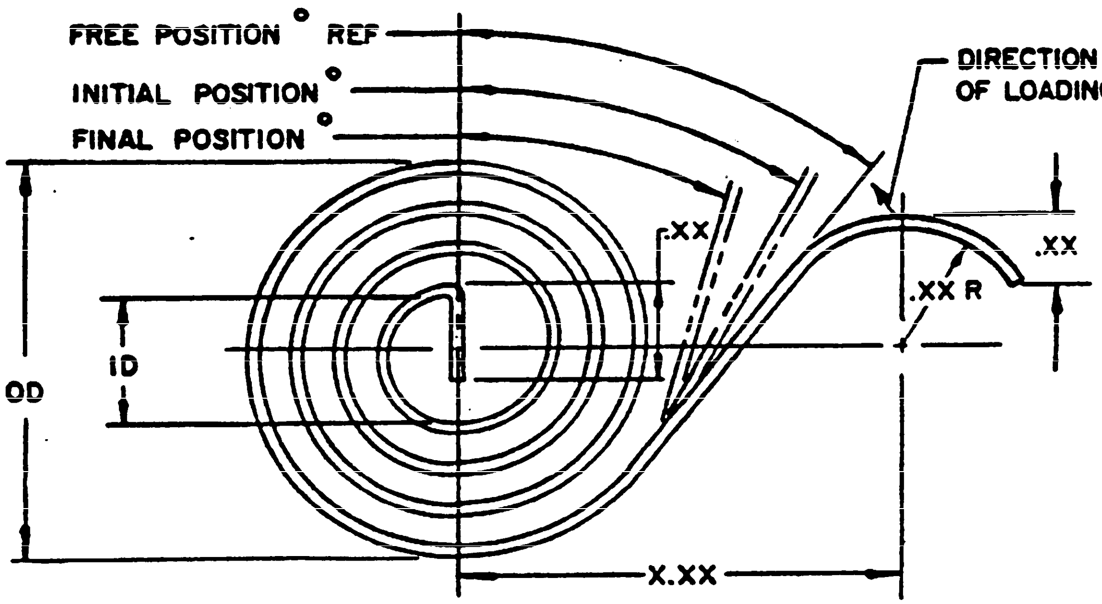
5.7.2.7 Type of Ends. Some of the types of ends applicable to helical torsion springs are illustrated in Figures 8 and 9. The types of ends required and the relative position of the ends shall be completely dimensioned and clearly delineated on the drawing.

5.8 SPIRAL TORSION SPRINGS

5.8.1 Definition. A spiral torsion spring is usually made by winding flat spring material on itself in the form of a spiral; and designed to wind up to exert pressure in a rotating direction around the spring axis. This pressure may be delivered as torque, or it may be converted into a push or pull force.

5.8.2 Drawing Requirements for Spiral Torsion Spring. Explanation of the spring data listed in Figure 10 is given below. Figure 23 is a typical detail drawing of a spiral torsion spring.

DRAWING REQUIREMENTS FOR SPIRAL TORSION SPRINGS



Spring data

		Para
MATERIAL SPECIFICATION		5.2.1
MATERIAL SIZE	XXX IN. THICK ... XX IN. WIDE	5.2.2
OUTSIDE DIAMETER IN FREE POSITION	X.XX IN. ± ___ XXX IN.	5.8.2.1
INSIDE DIAMETER	X.XX IN. ± ___ XXX IN.	5.8.2.1
DEVELOPED LG OF MATERIAL	X.XX IN. REF	5.8.2.2
ACTIVE LG OF MATERIAL	X.XX IN. REF	5.8.2.2
NUMBER OF COILS IN FREE POSITION	XX REF	5.8.2.3
TORQUE XX LB IN. ± XX LB IN. AT INITIAL POSITION		5.7.2.4
TORQUE XX LB IN. ± XX LB IN. AT FINAL POSITION		5.7.2.4
MAXIMUM DEFLECTION WITHOUT SET BEYOND FINAL POSITION	XX DEG	5.7.2.5
TYPE OF ENDS		5.8.2.4
SPECIAL DATA		5.3

FIGURE 10

TYPES OF SPIRAL TORSION SPRING ENDS

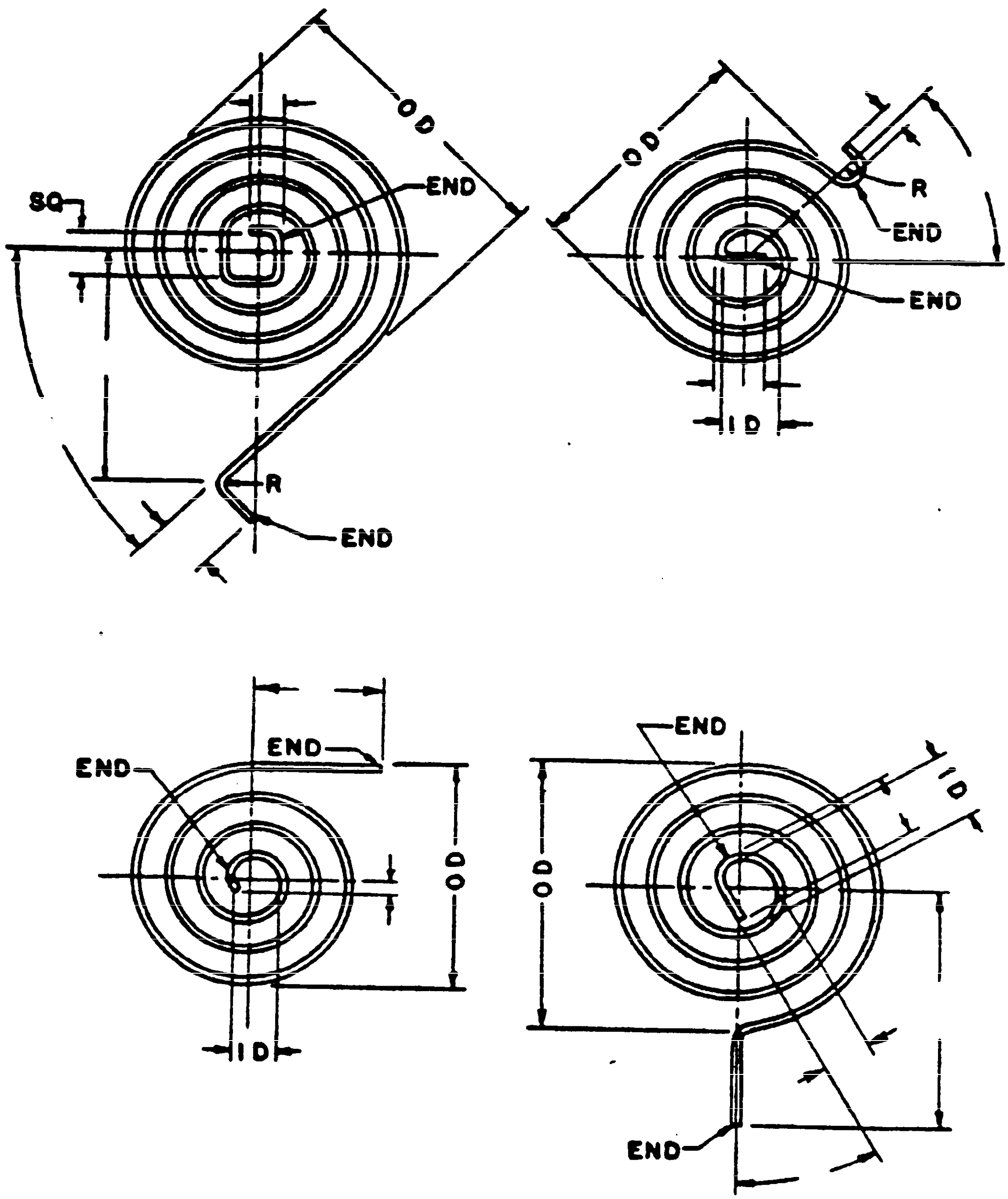


FIGURE 11

5.8.2.1 Outside and Inside Diameter of Coil in the Free Position. Specify both the outside and the inside diameters of the coil in the free position as toleranced dimensions.

5.8.2.2 Developed Length of Material and Active Length of Material. Specify the Developed and the Active Length of material as a REFERENCE. To the active length is added the inactive material forming the ends and the portion of a coil or coils that hug the shaft, the sum of which is the developed length.

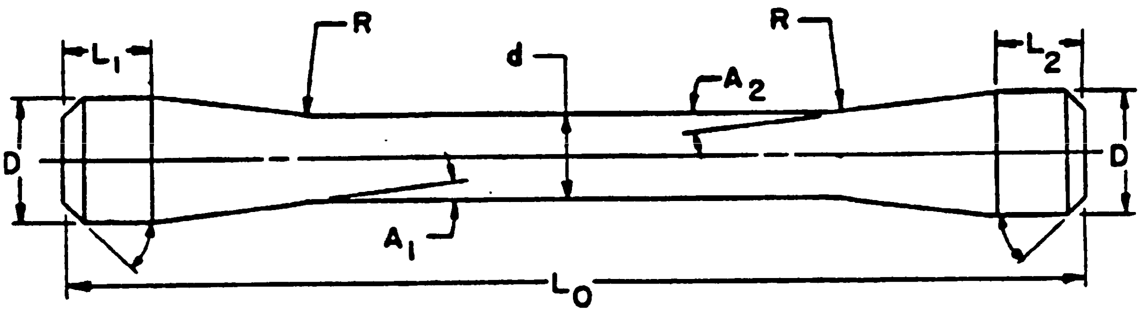
5.8.2.3 Number of Coils in Free Position. Specify and delineate the number of coils in the free position as a REFERENCE.

5.8.2.4 Type of Ends and Angular Relation of Ends. There are a variety of end formations possible for spiral springs, a few of which are illustrated in Figure 11. The ends shall be completely delineated and dimensioned on detail drawings.

5.9 TORSION BAR SPRINGS

5.9.1 Definition. Torsion bar springs are straight bars or rods of definite cross-section used to offer a resistance to a twisting moment about the longitudinal axis. The cross-section may be round, square, rectangular or hexagonal as dictated by design and availability. Circular cross-sections are generally used. Both ends are generally upset to di-

DRAWING REQUIREMENTS FOR TORSION BAR SPRINGS



Spring data

MATERIAL SPECIFICATION		5.2.1
DIAMETER OF BODY (d)	XX IN. ± XX IN.	5.9.2.1
DIAMETER OF ENDS (D)	XX IN. ± XX IN.	5.9.2.1
LG OF OVERALL (L_o)	XX IN. ± XX IN.	5.9.2.2
LG OF ENDS(S) (L_1 & L_2)	XX IN. ± XX IN.	5.9.2.2
ANGLE OF TAPER(S) (A_1 & A_2)	XX DEG ± XX DEG	5.9.2.3
TORQUE ...... LB IN. ± ...... LB IN. AT ...... DEGREES DEFLECTION		5.9.2.4
SPRING RATE	XX LB IN./DEG ...... REF	5.9.2.5
TYPE OF ENDS		5.9.2.6
DIRECTION OF WINDUP MARKING		5.9.2.7
AANGULAR RELATION BETWEEN ENDS		5.9.2.8
SPECIAL DATA (SAE MANUAL ON DESIGN AND MANUFACTURE OF TORSION BARS)		Fig. 25

FIGURE 12

ameters larger than the body diameters with a tapered transition section between body and ends to keep stress concentrations to a minimum. Splines, serrations, or other types of couplings are cut on the upset ends to form the means of anchorage.

5.9.2 Drawing Requirements for Torsion Bar Springs. Explanation of the spring data listed in Figure 12 is given below. Figure 24 is a typical detail drawing of a torsion bar spring, and Figure 25 lists processing data.

5.9.2.1 Diameters of Body and Ends. Specify in the delineation both the end diameter and the body diameter as toleranced dimensions.

5.9.2.2 Length Overall and of Ends. Specify in the front view the overall length, and the length of ends, with tolerances.

5.9.2.3 Angle of Taper. Specify the angle of taper with a tolerance in the front view.

5.9.2.4 Torque Loads. The torque loads with tolerances should be specified in pound inches or pound feet, as appropriate, at definite amounts of deflection in degrees of rotation from the free position. Torque loads should preferably be specified at the initial and the final operating positions in the assembly.

5.9.2.5 Spring Rate. Specify the spring rate in lb in./degree or lb ft/degree, as a REFERENCE.

5.9.2.6 Type of Ends. The type of end (involute spline, serrations, etc.) shall be completely delineated and dimensioned on detail drawings.

5.9.2.7 Direction of Windup Marking. For coldset torsion bar springs, the direction of windup (load application in service) shall be indicated with an arrow on the end of the spring. For torsion bar springs which are not coldset, direction of windup marking is not required.

5.9.2.8 Angular Relation between Ends. For torsion bar springs which are coldset a scribe line (approximately $\frac{1}{16}$ x 60°V) shall be cut on each end of the spring. The end view shall show the angular relation between the ends before and after coldsetting. For torsion bar springs which are not coldset, angular relation between ends is not required unless an angular relation must be maintained between involute spline or serration teeth.

5.10 VOLUTE SPRINGS

5.10.1 Definition. Volute Springs are conical shaped compression springs made of rectangular cross-sectional material so constructed that the coils are capable of telescoping into each other.

5.10.2 Drawing Requirements for Volute Springs. Explanation of the spring data listed in Figure 13 is given below. Figure 26 is a typical detail drawing of a volute spring.

5.10.2.1 Outside and inside Diameter. Specify in the delineation both the outside and the inside diameters as toleranced dimensions.

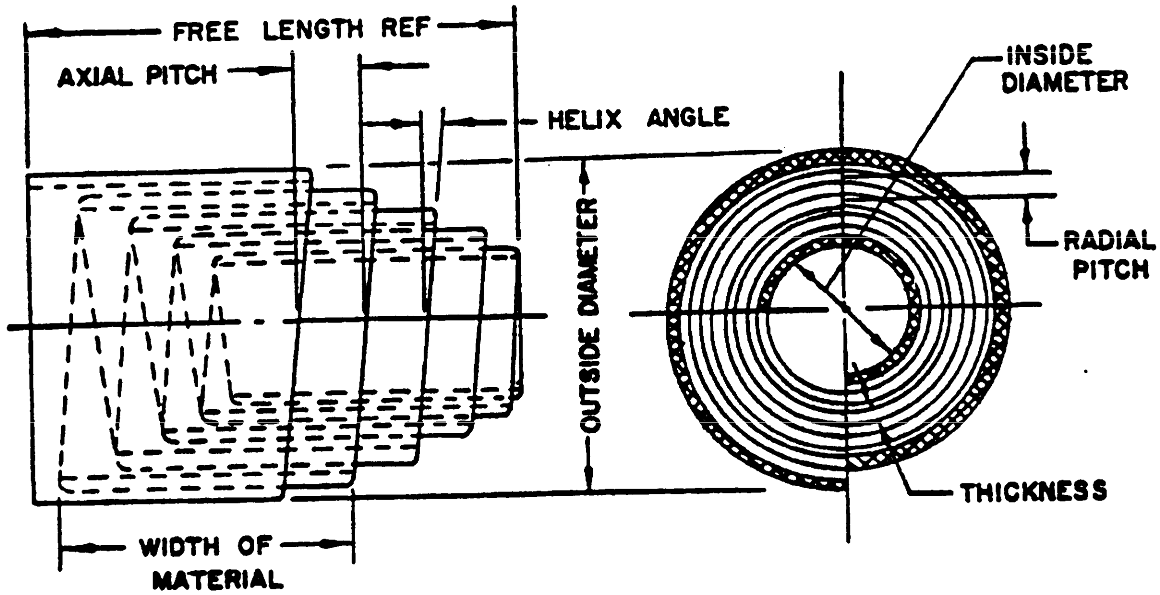
5.10.2.2 Free Length. Specify the free length as a REFERENCE dimension in the delineation.

5.10.2.3 Total Coils. Specify the number of total coils, as a REFERENCE, or with a tolerance if essential to the design.

5.10.2.4 Radial Pitch. Specify the radial pitch in inches with a tolerance.

5.10.2.5 Axial Pitch and Helix Angle. The axial pitch and the helix angle of a volute spring determine its loads, length and bottoming characteristics. Volute springs with a constant axial pitch will have the largest active coil bottoming first. A spring with a constantly reducing axial pitch will have a smaller pitch for the smaller coils and such springs can be designed so that all active

DRAWING REQUIREMENTS FOR VOLUTE SPRINGS



Spring data

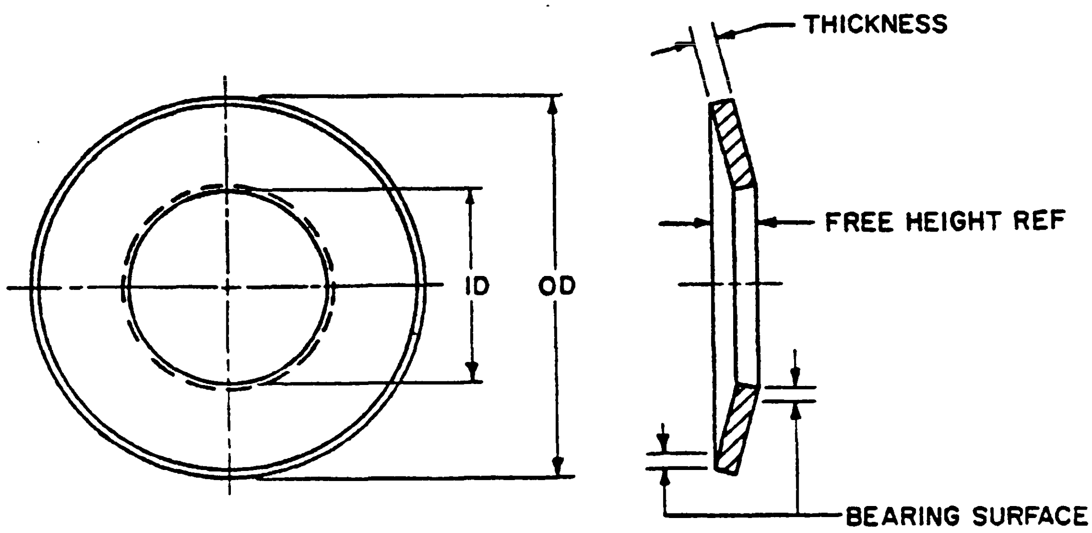
		Para
MATERIAL SPECIFICATION		5.2.1
MATERIAL SIZE	XXX IN. THICK ... XXX IN. WIDE	5.2.2
OUTSIDE DIAMETER OF COIL	XXX IN. ± ... XX IN.	5.10.2.1
INSIDE DIAMETER OF COIL	XXX IN. ± ... XX IN.	5.10.2.1
FREE LG	XXX IN. REF	5.10.2.2
TOTAL COILS		5.10.2.3
DIRECTION OF HELIX		5.4.2.4
RADIAL PITCH	XX IN. ± ... XX IN.	5.10.2.4
AXIAL PITCH	XX IN. ± ... XX IN.	5.10.2.5
HELIX ANGLE	XX DEG ± ... XX DEG	5.10.2.5
LOAD AT COMPRESSED LG OF	XX IN.	
—	XX LB ± ... XXX LB	
LOAD AT COMPRESSED LG OF	XX IN.	5.10.2.6
—	XXLB ± ... XX LB	5.10.2.6
SOLID LG	XX IN. MAX	5.10.2.7
SPECIAL DATA	(MIL-S-13475 AND SAE MANUAL ON DESIGN AND MANUFACTURE OF VOLUTE SPRINGS)	5.3

FIGURE 13

coils theoretically bottom at about the same time. Specify either the helix angle or preferably the axial pitch with a tolerance as constant. If the helix angle is constant the pitch will vary; if the pitch is constant, the helix angle will vary. It is impossible because of the nature of the spring to have both the helix angle and the pitch as constant. However, it is possible for both pitch and helix angle to vary throughout a volute spring.

5.10.2.6 Loads. Specify the loads with tolerances to be developed at definite compressed lengths, preferably at the initial and the final operating positions in the assembly.

DRAWING REQUIREMENTS FOR CONED DISC (BELLEVILLE) SPRINGS



Spring data

	Para
MATERIAL SPECIFICATION	5.2.1
THICKNESS OF MATERIAL .... XXX IN.	5.2.2
INSIDE DIAMETER .... XX IN. ± .... XX IN.	5.11.2.1
OUTSIDE DIAMETER .... XX IN. ± .... XXX IN.	5.11.2.1
METHOD OF STACKING	5.11.2.2
FREE HEIGHT .... XXX IN. REF	5.11.2.2
LOAD AT COMPRESSED HEIGHT OF .... XXX IN. – .... XX LB ± .... XX LB WHEN STACKED IN .... (OR FOR INDIVIDUAL DISCS)	5.11.2.3
LOAD AT COMPRESSED HEIGHT OF .... XXX IN. – .... XX LB ± .... XX LB WHEN STACKED IN	5.11.2.3
BEARING SURFACES .... XXX IN. MAX	5.11.2.4
BEARING SURFACES MUST BE PARALLEL WITHIN .... XXX IN. AT .... XXX IN. FROM CENTER	5.11.2.5
O D AND I D SHALL BE CONCENTRIC WITHIN .... XXX TIR	5.11.2.6
SPECIAL DATA (MIL-S-12133)	5.3

FIGURE 14

5.10.2.7 Solid Length. Specify the solid length as a maximum dimension allowing for the tolerance on the width of the blade, protective coating, etc. The spring should not ordinarily be permitted to go solid in operation, except when used as a bumper or dictated by other design requirements.

5.11 CONED DISC (BELLEVILLE) SPRINGS

5.11.1 Definition. A coned disc (Belleville) spring is a spring washer in the form of a frustum of a cone, having constant material thickness, and used as a compression spring.

5.11.2 Drawing Requirements for Coned Disc (Belleville) Springs. Explanation of the spring data listed in Figure 14 is given below. Figure 27 is a typical detail drawing of a coned disc (Belleville) spring.

METHODS OF STACKING CONED DISC (BELLEVILLE) SPRINGS

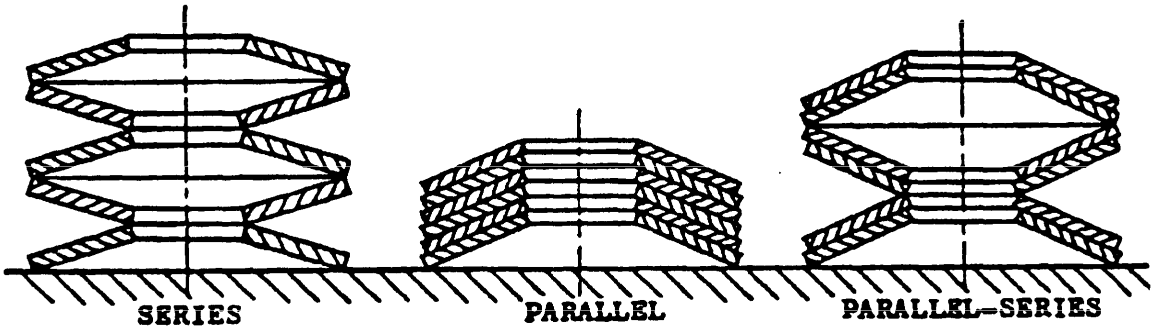


FIGURE 15

5.11.2.1 Inside and Outside Diameters. Speciify both the inside and outside diameters, as toleranced dimensions, preferably in the delineation.

5.11.2.2 Free Height and Method of Stacking. It is usually best to specify the free height of the individual disc as a REFERENCE dimension, but if essential to the design it may be a toleranced dimension. When springs are used in sets, as in Figure 15, show on an assembly drawing the method of stacking and specify the free height of the stack as a REFERENCE dimension in the delineation. However, on some applications where the free height is critical, a tolerance may be added to the stacked height.

5.11.2.3 Loads. When discs are to be installed singularly, only one load with a tolerance applied shall be specified. When discs are to be installed in sets, specify the loads with a tolerance at definite compressed lengths, preferably at the initial and final operating positions of the mechanism.

When discs are used in sets, a load for individual discs shall not be specified. When discs are used in sets, the following note shall be included on detail drawings: STACKS SHALL BE SECURED TOGETHER IN THE SEQUENCE OF TESTING AND SUCH SEQUENCE SHALL BE RETAINED UPON INSTALLATION.

5.11.2.4 Bearing Surfaces. Bearing surfaces may be provided when necessary to meet the design requirements. They aid in meeting tests for loads, tolerances and burr removal. They should not exceed .080 inches wide.

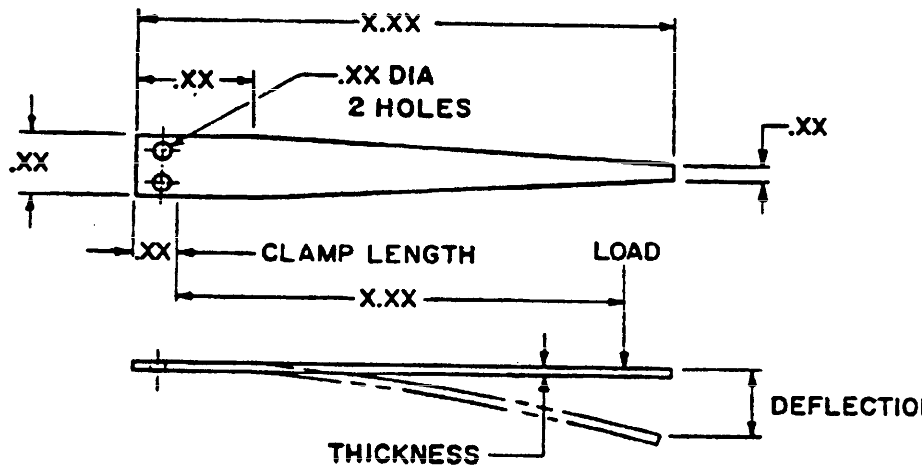
5.11.2.5 Bearing Surfaces Parallel. Such surfaces must be parallel in an amount equal to or less than 15% of the Free Height, when measured at a distance of twice the Outside Diameter, located from the center line.

5.11.2.6 Concentricity between Outside and Inside Diameters. Specify the concentricity between the outside and inside diameters when important. When specified, the outside diameter of the springs shall be concentric with the inside diameter within .003 inches (.006 TIR) for springs 2 inches and less in outside diameter, and .005 inches (.010 TIR) for springs over 2 inches in outside diameter.

5.1 FLAT SPRINGS

5.12.1 Definition. A flat spring is in a broad sense any spring made of flat strip or bar stock, which deflects as a cantilever or as a simple beam. Any metal stamping which stores energy when deflected and returns an equal amount of energy is a flat spring. Ex-

DRAWING REQUIREMENTS FOR FLAT SPRINGS



Spring data

		Para
MATERIAL SPECIFICATION		5.2.1
MATERIAL SIZE	X.XX IN. THICK .... XX IN. WIDE	5.2.2
DEVELOPED (OVERALL) LG	X.XX IN. REF	5.12.2.1
CLAMP LG	XX IN. ± .... XX IN.	5.12.2.2
LOAD AT X.XX IN. DEFLECTION – XX LB ± XX LB		5.12.2.3
LOAD AT X.XX IN. DEFLECTION – XX LB ± XX LB		5.12.2.3
SPECIAL DATA		5.3

FIGURE 16

cluded from this definition are Coned Disc (Belleville) Springs, Spiral Torsion Springs, Spring Washers and Rings.

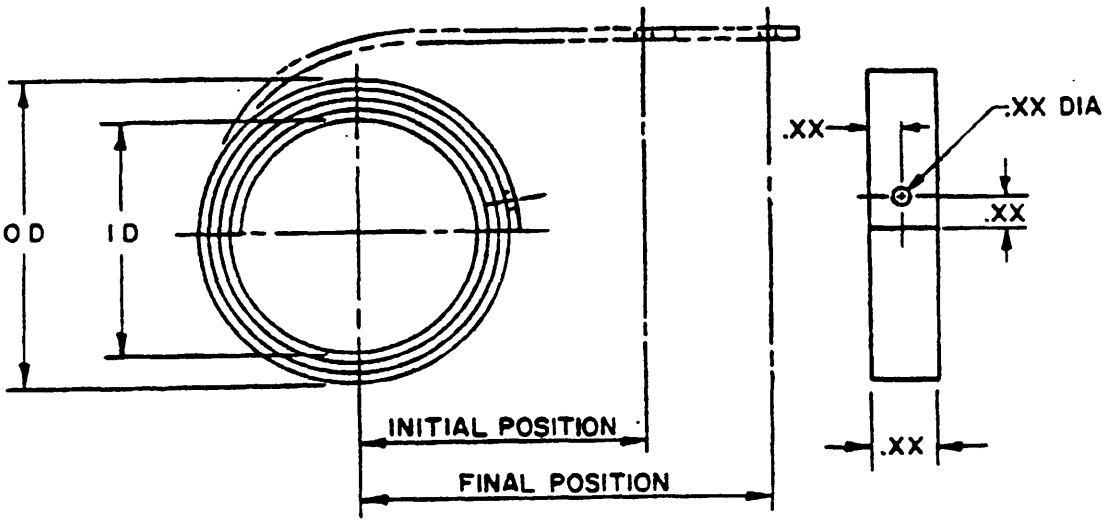
5.12.2 Drawing Requirements for Flat Springs. Explanation of the spring data listed in Figure 16 is given below. Figure 28 is a typical detail drawing of a flat spring. Because of the wide variation of shapes for flat springs a complete specification is beyond the scope of this standard.

5.12.2.1 Developed Lengths. Specify the developed overall length as a REFERENCE or with a tolerance if essential to keep within space limitations.

5.12.2.2 Clamp Length. Specify the clamp length with a tolerance. The clamp length is used for mounting and load testing thereby rendering that portion inactive.

5.12.2.3 Loads. Specify load at distance from end of clamp length. This distance is the moment arm of the load. Indicate the direction of the applied load to produce a prescribed deflection. When applicable, loads and moments at the initial and final operating positions shall be specified. Tolerances shall be applied to loads not to deflections. When necessary for purposes of clarity, a deflected position as shown in Figure 28 may be used.

DRAWING REQUIREMENTS FOR CONSTANT FORCE SPRINGS



Spring data

				Para
MATERIAL SPECIFICATION				5.2.1
MATERIAL SIZE	XXX IN. THICK		XX IN. WIDE	5.2.2
OUTSIDE DIAMETER	XX IN. ±		XX IN.	5.8.2.1
INSIDE DIAMETER	XX IN. ±		XX IN.	5.8.2.1
DEVELOPED LG			XX IN. REF	5.8.2.2
ACTIVE LG			XX IN. REF	5.8.2.2
NUMBER OF COILS			X REF	5.8.2.3
LOAD	XX LB ±		XX LB (CONSTANTLY)	5.13.2.1
INITIAL POSITION			X IN.	5.13.2.2
FINAL POSITION			XX IN.	5.13.2.2
TYPE OF ENDS				5.13.2.3
FITS OVER			XXX DIA ROLLER	5.13.2.4

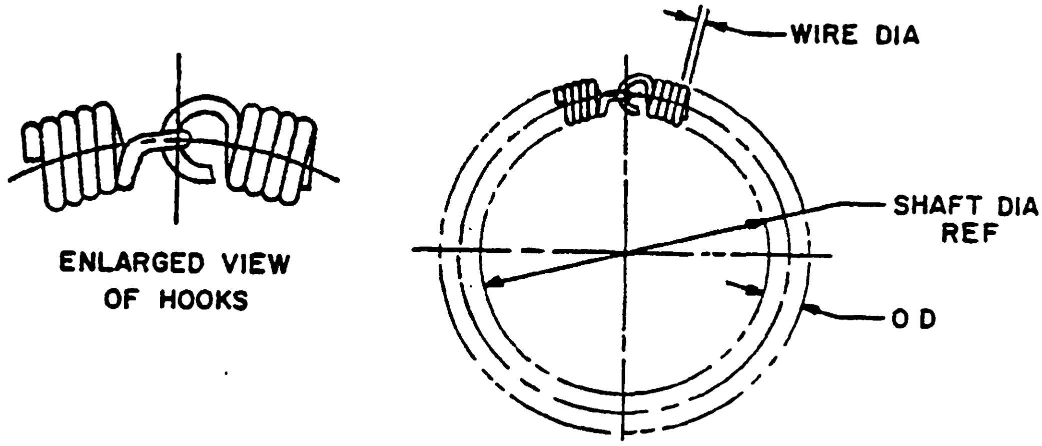
FIGURE 17

5.13 CONSTANT FORCE SPRINGS

5.13.1 Definition. A constant force spring is one that is made from strip material, similar to a clock or motor spring except that the inner end usually is not fastened thus leaving it free to rotate and the coils are formed to remain in a coiled position without expanding. Retaining rings to hold the coils are not required. Such a spring can have its outer end pulled outwardly and the force exerted to uncoil the spring can be uniformly constant with each inch of deflection. Various types and sizes can be made.

5.13.2 Drawing Requirements for Constant Force Springs. Explanation of the spring data listed in Figure 17 is given below. Figure 29 is a typical detail drawing of a constant force spring.

DRAWING REQUIREMENTS FOR GARTER SPRINGS



Spring dtata

SPECIFY SPRING DATA SHOWN ON FIGURES 4(a) THRU 4(d), AS APPLICABLE.
ADDITIONALLY, THE FOLLOWING CHARACTERISTICS SHALL BE SPECIFIED:

		Para
SHAFT DIAMETER	XX IN. REF	5.14.2.1
COMPRESSIVE LOAD PER INCH OF CIRCUMFERENCE ON SHAFT	XX LB. REF	5.14.2.2
SPECIAL DATA		5.3

FIGURE 18

5.13.2.1 **Loads.** Specify the load in pounds with a tolerance. This load should remain uniformly constant.

5.13.2.2 **Initial Position and Final Position.** Specify the initial position and final position in inches without a tolerance.

5.13.2.3 **Type of Ends.** A large variety of end formations are possible. Both ends can be cut at a radius approximately equal to the width of the material. The inner end usually is not punched or formed as it must be free to rotate over a pulley or roller. The outer end may have a punched hole or slot or be formed to fit into a holding device for fastening purposes. The ends should be completely delineated on the drawing.

5.13.2.4 **Fits over Roller.** Most constant force springs are intended for use over rollers and they should fit snugly on them. The roller diameter is usually 15 to 20% larger than the inside diameter of the spring and should be specified on the drawing.

5.13.2.5 **Thickness.** The thickness of the material is usually so thin that it is desirable to exaggerate the thickness on drawings for illustrative purposes. Also, the exact number of coils need not be shown in the drawing.

5.14 GARTER SPRINGS

5.14.1 Definition. A garter spring is a special close coiled, long, extension spring with its ends fastened together and used in the form of a ring. Such springs are used principally in mechanical seals on shafting; to hold round segments together; as belts and as holding devices. It is customary to order the springs in straight lengths and fasten the ends at assembly.

5.14.2 Drawing Requirements for Garter Springs. Explanation of the spring data listed in Figure 18 is given below. Figure 30 is a typical detail drawing of a garter spring.

5.14.2.1 Shaft Diameter. Specify the shaft diameter over which the spring works as a REFERENCE dimension.

5.14.2.2 Compressive Load per Inch of Circumference on Shaft. Specify this load as a REFERENCE only if essential to the design.

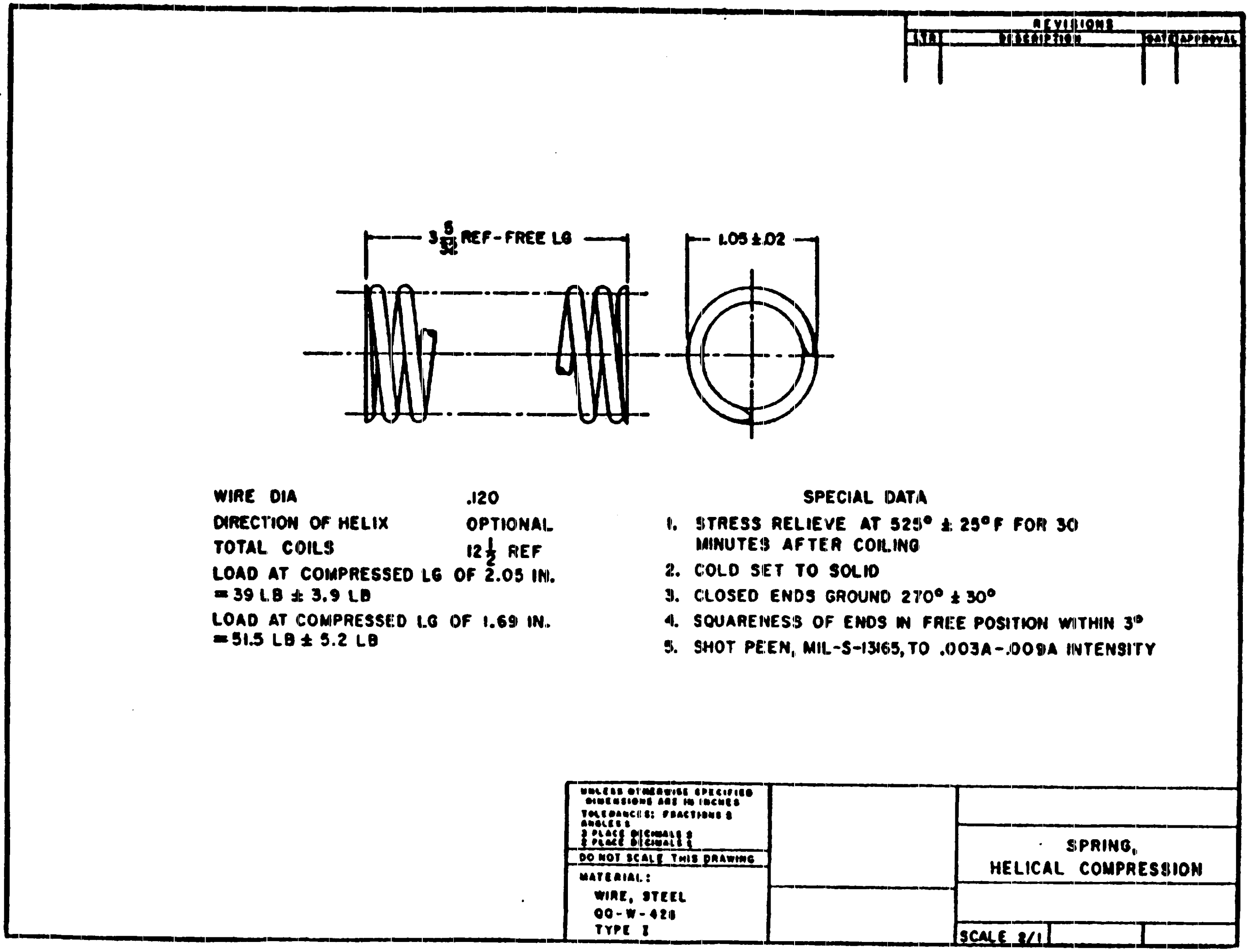


FIGURE 19. *Typical detail drawing of compression spring*

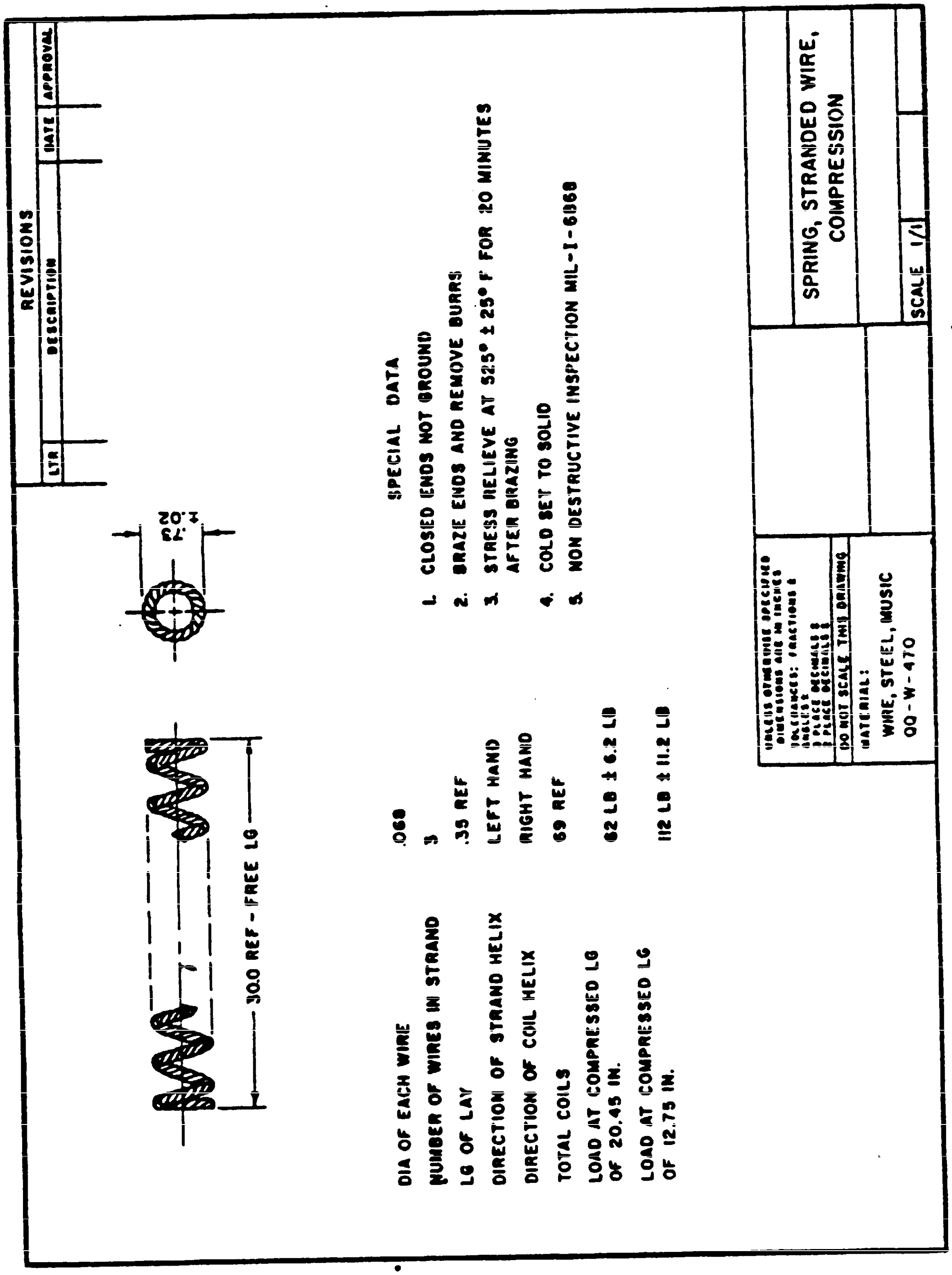


FIGURE 20. *Typical detail drawing of stranded wire compression spring*

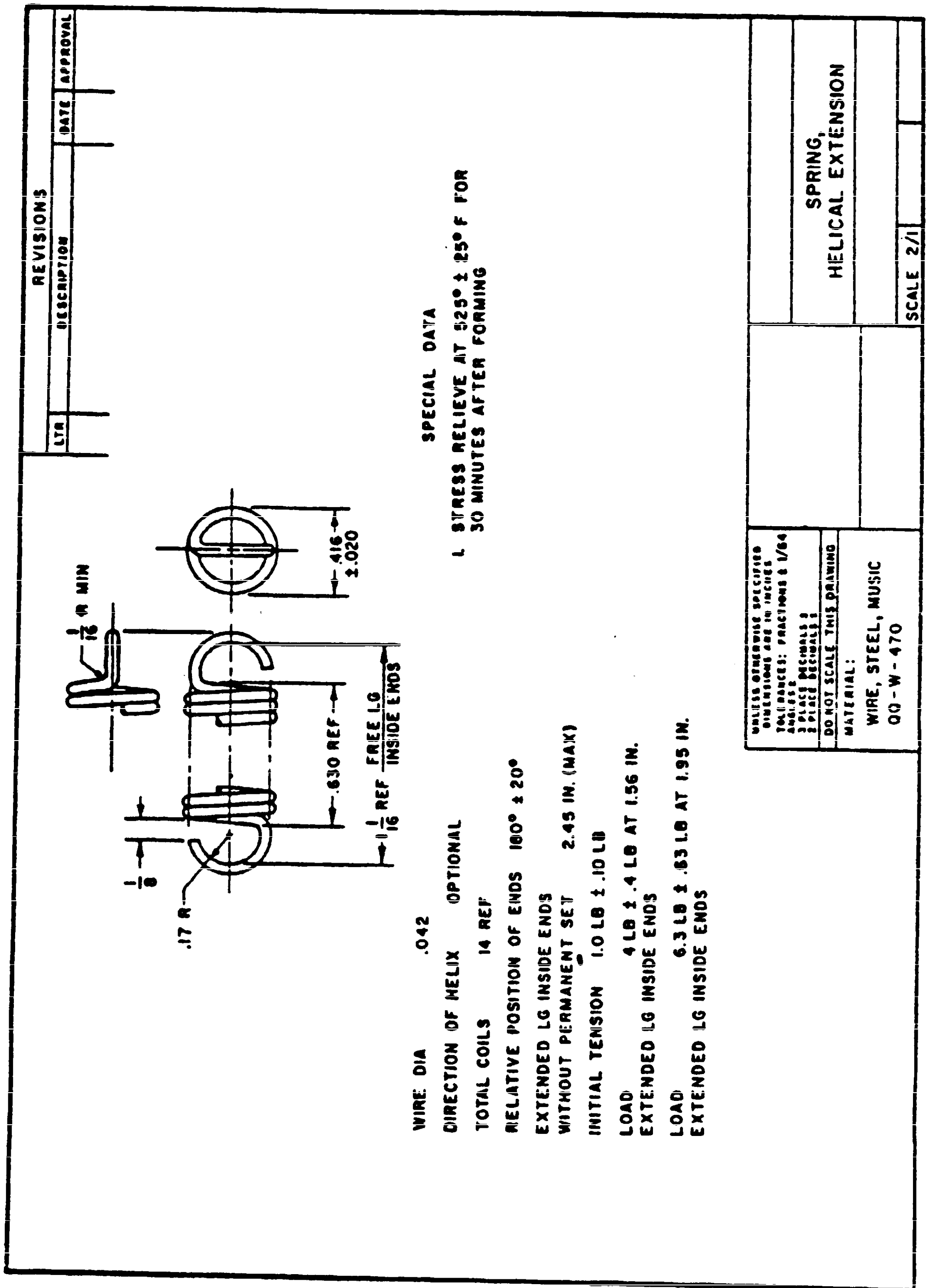


FIGURE 21. *Typical detail drawing of extension spring*

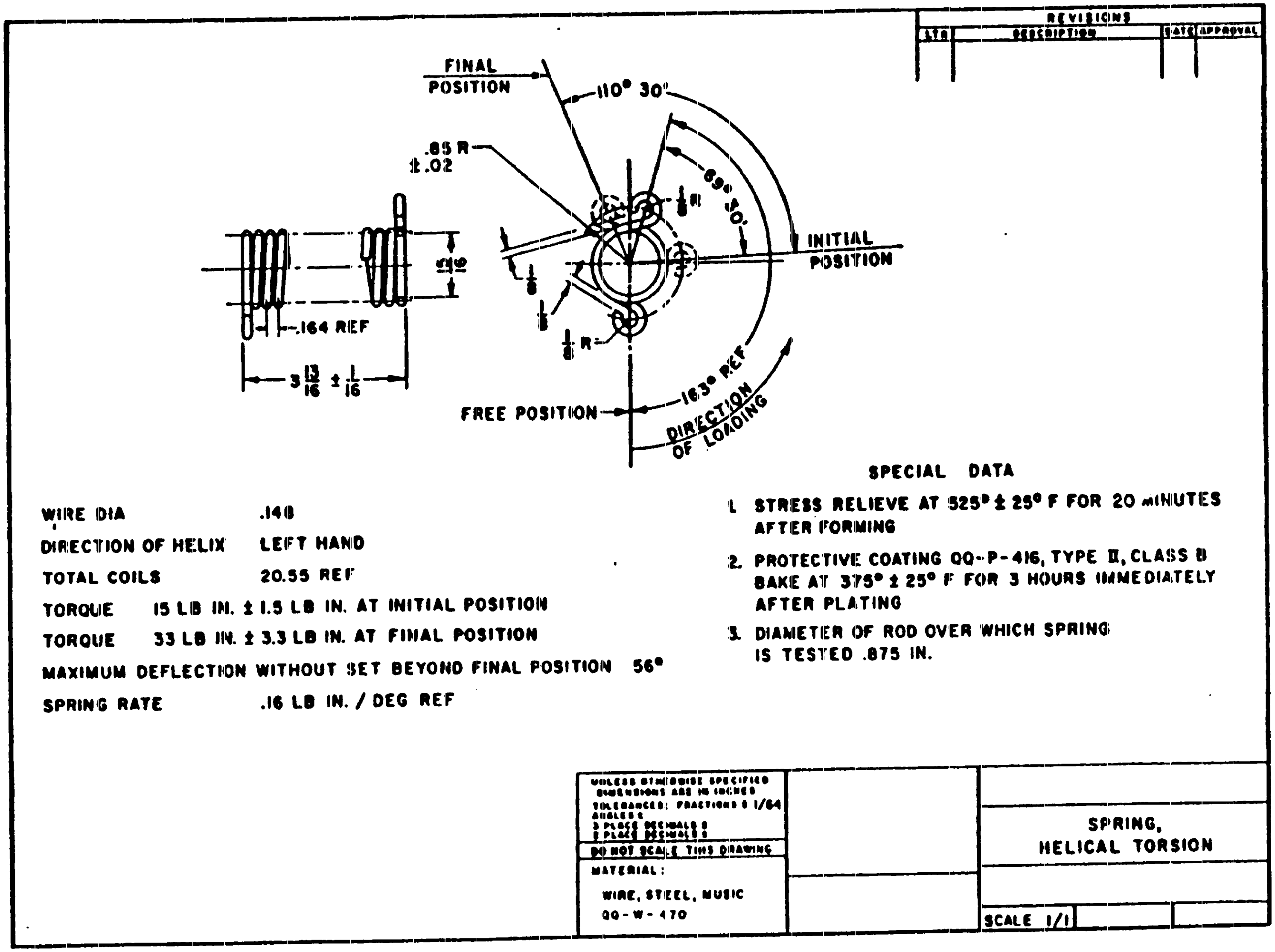


FIGURE 22. *Typical detail drawing of helical torsion spring*

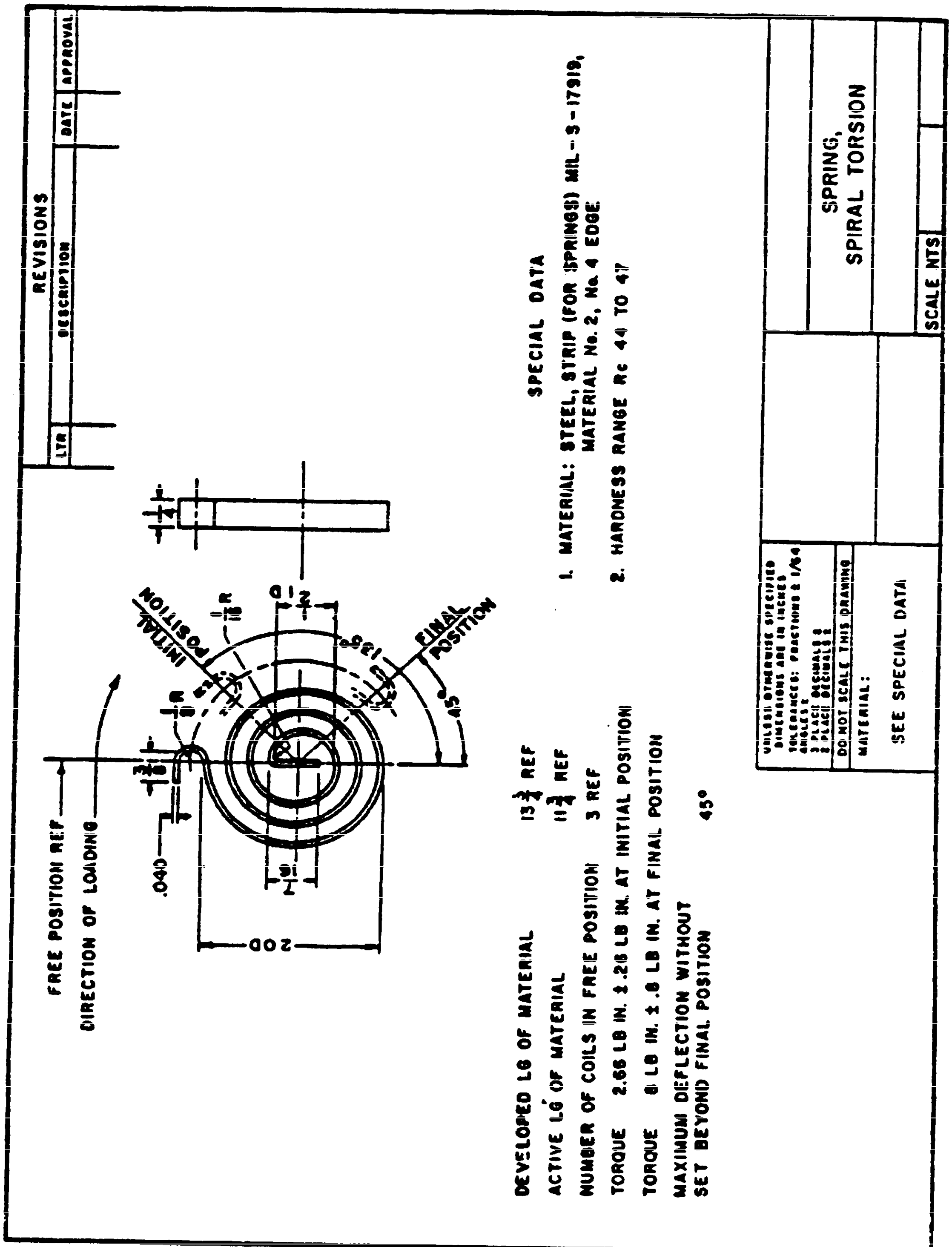


FIGURE 23. *Typical detail drawing of spiral torsion spring*

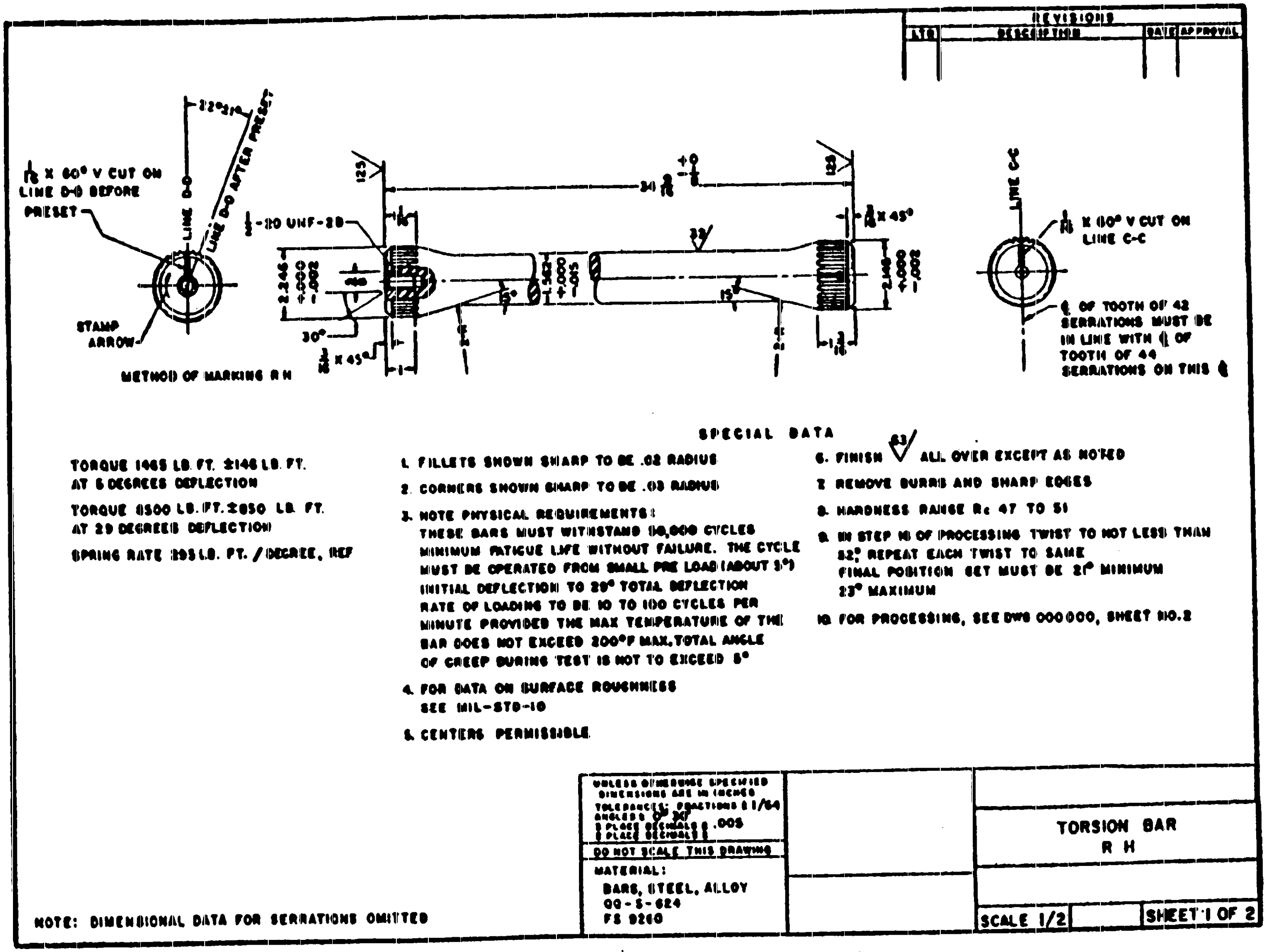


FIGURE 24. *Typical detail drawing of torsion bar spring*

PROCESSING FOR TORSION BAR (PERFORM IN THE SEQUENCE INDICATED BELOW)

1. Upset ends.
2. Cycle anneal (Brinnell Hardness 207-283).
3. Clean (grit or pickle).
4. Straighten.
5. Center.
6. Face ends.
7. Drill and tap threaded hole.
8. Finish entire surface of the spring to dimensional limits.

9A. Stamp or electric etch part number and arrow on end of bar. Note: The order in which operations 2 through 9 is performed is optional with the manufacturer.

9B. Operation 15 may be performed prior to operation 10 at the option of the manufacturer.

10. Heat treat as follows:
   A. Heat (in controlled atmosphere furnace if necessary to control decarburization) and quench in hot oil (125° - 150° F).
   B. Temper to hardness specified on part drawing.
   C. The quenched Rockwell "C" hardness is to be checked periodically to maintain control over the extent of decarburization occurring from the quenching operation.

11. Straighten hot-off-tempering heat. No straightening to be done below 500° F. Leave bars in holding furnace at 500° - 700°, if necessary, to await straightening. Check temperature during straightening with tempil stik or pyrometric cone.

12. Check final hardness on surface of upset diameter after finished heat treatment and before hobbing serrations.

13. Magnetic particle inspection as follows:
   A. Magnetize the torsion bar between the contact posts, using the maximum rated current strength that can be developed on the machine (approximately 4,000 amperes). (And using not more than 3 "shots" of current), and apply the indicating solution while the magnetizing current is flowing. If no indications of defects appear or are observable, the part is acceptable.
   B. If indication of defects appear, wipe the surface clean of all indications and apply the indicating solution again without the current flowing. If the indications do not reappear, it indicates that the defects are not serious and the part may be accepted.
   C. If the indications reappear, demagnetize, magnetize again as in "A" (except use a current strength of 1,000 amperes per inch of body diameter), (and using not more than 3 "shots" of current), and apply the indicating solution, without the current flowing. If the indications do not appear at this current strength, the defects are of a minor nature and the part may be acceptable. If the indications appear at the current strength, the part should be rejected.
   D. Demagnetize.
   E. If acceptable, electric etch the letter "M" on outer end.

14. Shot peen body with .035 to 0.43 diameter rounded steel cut wire shot to an intensity of .012A2 to .015A2, Spec. MIL-S-13165, 100% visual coverage. Shot to be Spec. MIL-S-16410 rounded steel cut wire, hard drawn or oil quenched and tempered to Rockwell "C" 46-54. If clipped wire shot is not readily available, cast steel shot of same diameter, but with a hardness of Rockwell "C" 50-60, may be used until clipped wire shot is available.

15. Hob serrations, see note 9B.

16. Pre-set as follows:
   A. Anchor one end and twist other end 3 to 5 consecutive times to degree specified on applicable part drawing. If the spring takes a set equal to that specified on applicable part drawing, the spring is acceptable. If at any time during the presetting the spring takes a set other than that specified on applicable part drawing, it should be rejected.

   Note: Twist clockwise for spring marked ↻ and counter-clockwise for springs marked ↺ never twist or work a spring against the direction indicated by arrow stamped on it.

17. Machine one serration from each end as per applicable part drawing.

18. Mill, stamp or grind indexing notch on trailing end.

19. Shot peen serrations.
   Using .023 - 0.28 diameter rounded steel cut wire shot to an intensity of .012A2 to .015A2, Spec. MIL-S-13165, 100% visual coverage. Shot to be Spec. MIL-S-16410 rounded steel cut wire, hard drawn or oil quenched and tempered to Rockwell "C" 46-51. If clipped wire shot is not readily available, cast steel shot of same diameter, but with a hardness of Rockwell "C" 50-60, may be used until clipped wire shot is available.

20. Demagnetize, if necessary, to remove any residual magnetism remaining from pre-set or shot peening operation.

21. Clean surface of bar thoroughly by any applicable method to remove all grease, oil, dirt, etc.

22. Support bar at serrated ends, and brush, spray or spread approved adhesives all over except serrations and tapped hole, to obtain a coating thickness of 0.10 inch minimum. Allow adhesive to dry thoroughly with bar supported at serrated ends.

For Torsion Bar Detail, See DWG 000 000, Sheet No. 1

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONS ± ANGLES ± 3 PLACE DECIMALS ± 2 PLACE DECIMALS ±		
DO NOT SCALE THIS DRAWING		TORSION BAR PROCESSING
MATERIAL:		SCALE SHEET 2

FIGURE 25

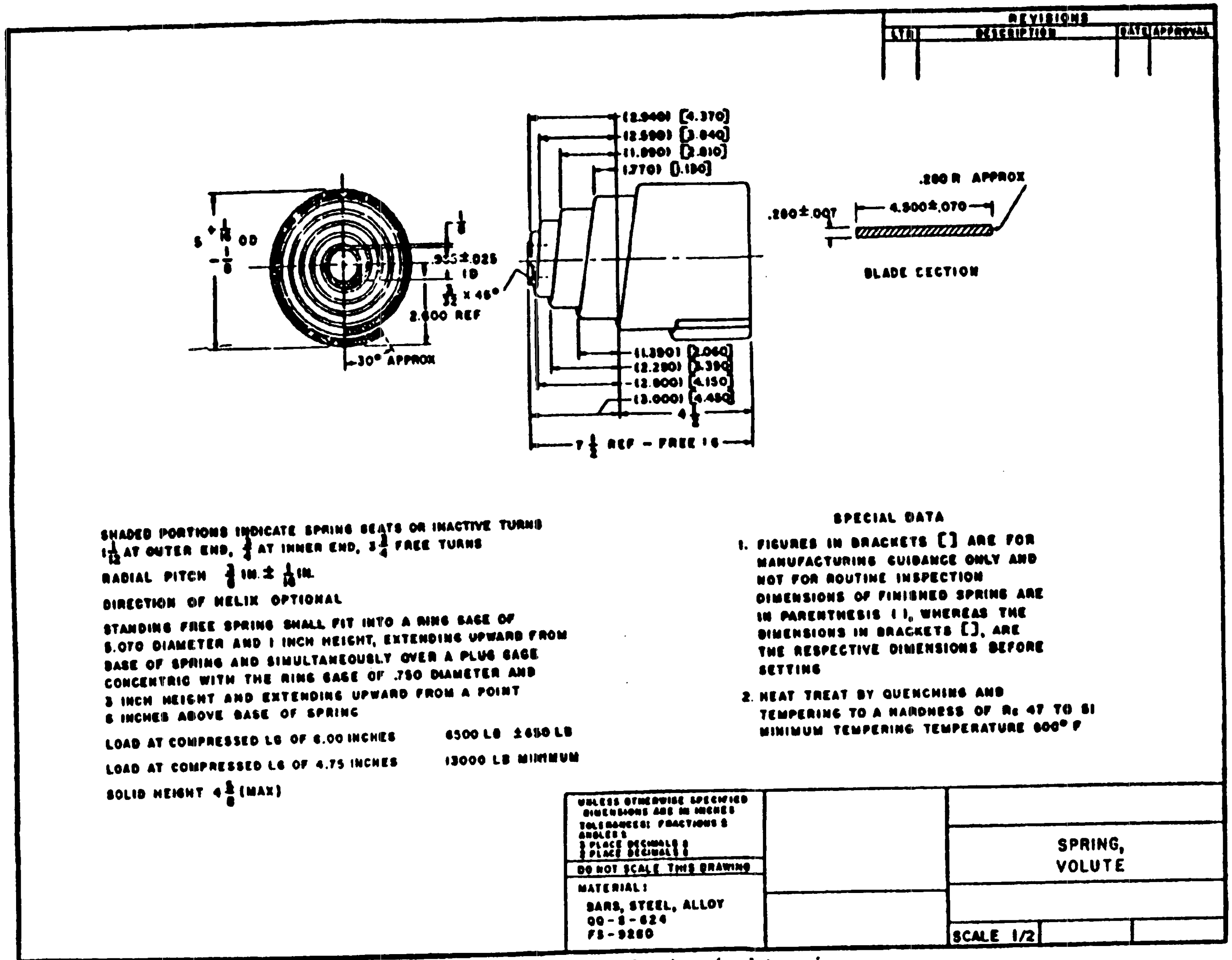


FIGURE 26. *Typical detail drawing of volute spring*

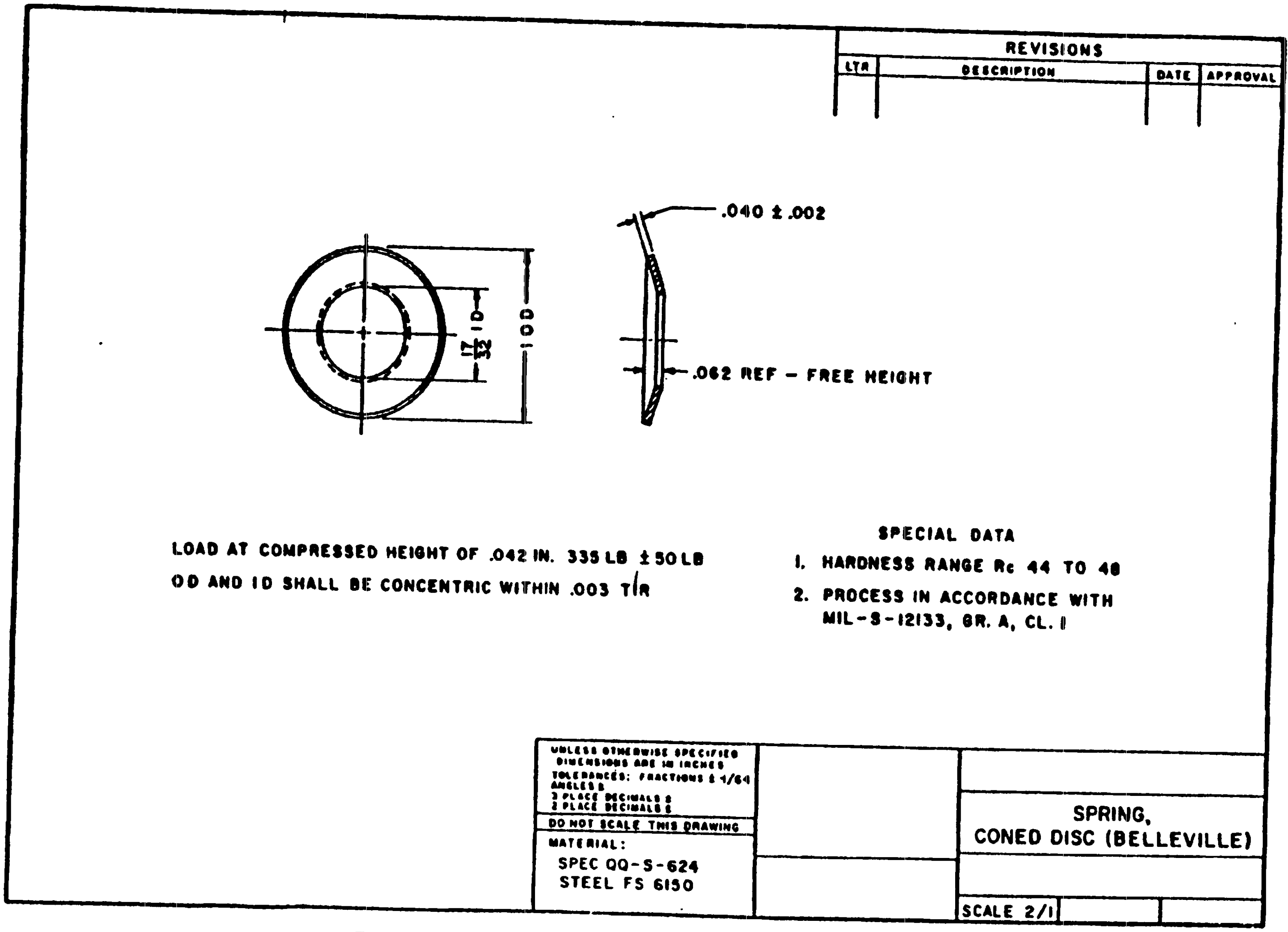


FIGURE 27. *Typical detail drawing of coned disc (Belleville) spring*

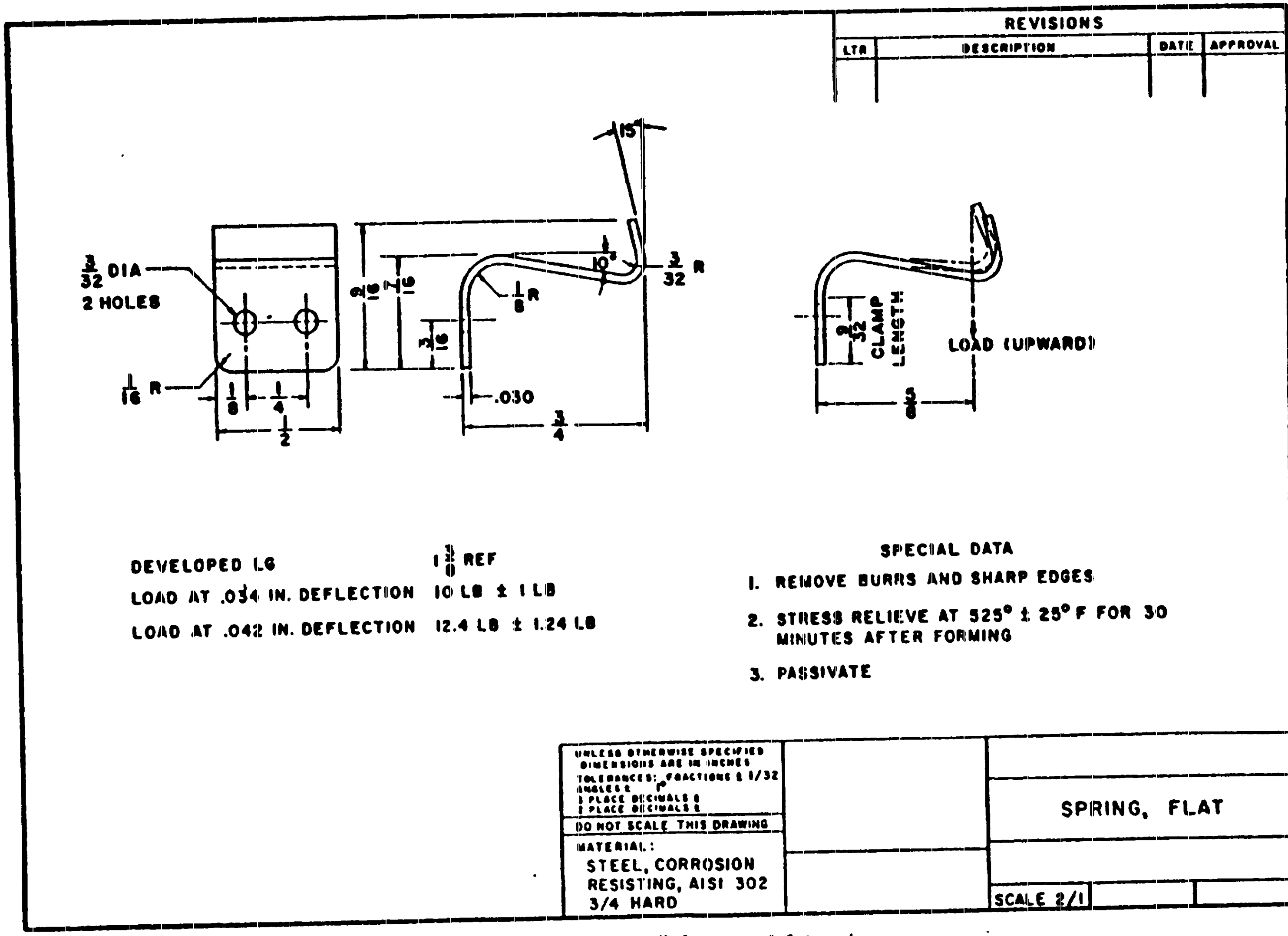


FIGURE 28. *Typical detail drawing of flat spring*

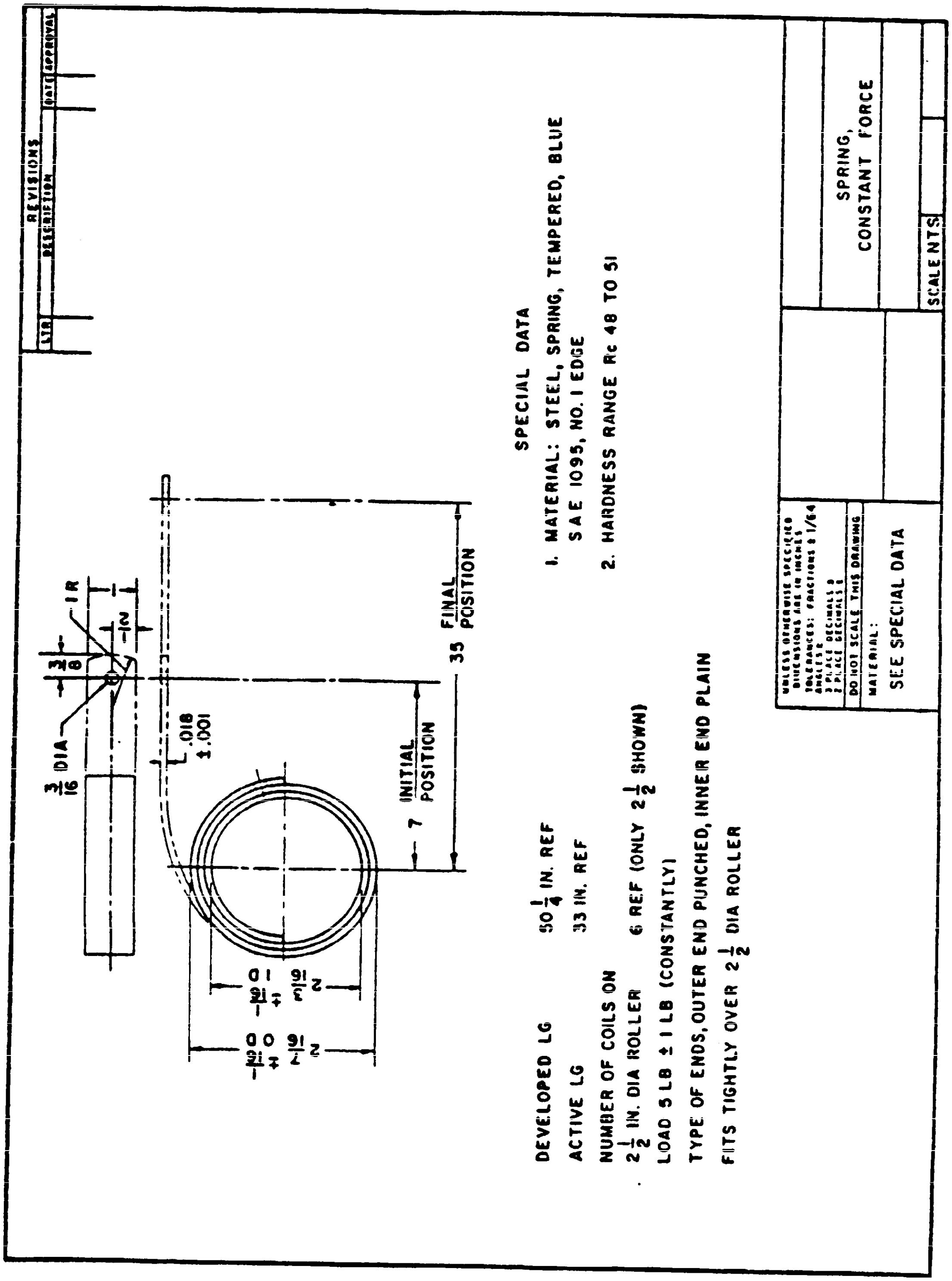


FIGURE 29. *Typical detail drawing of constant force spring*

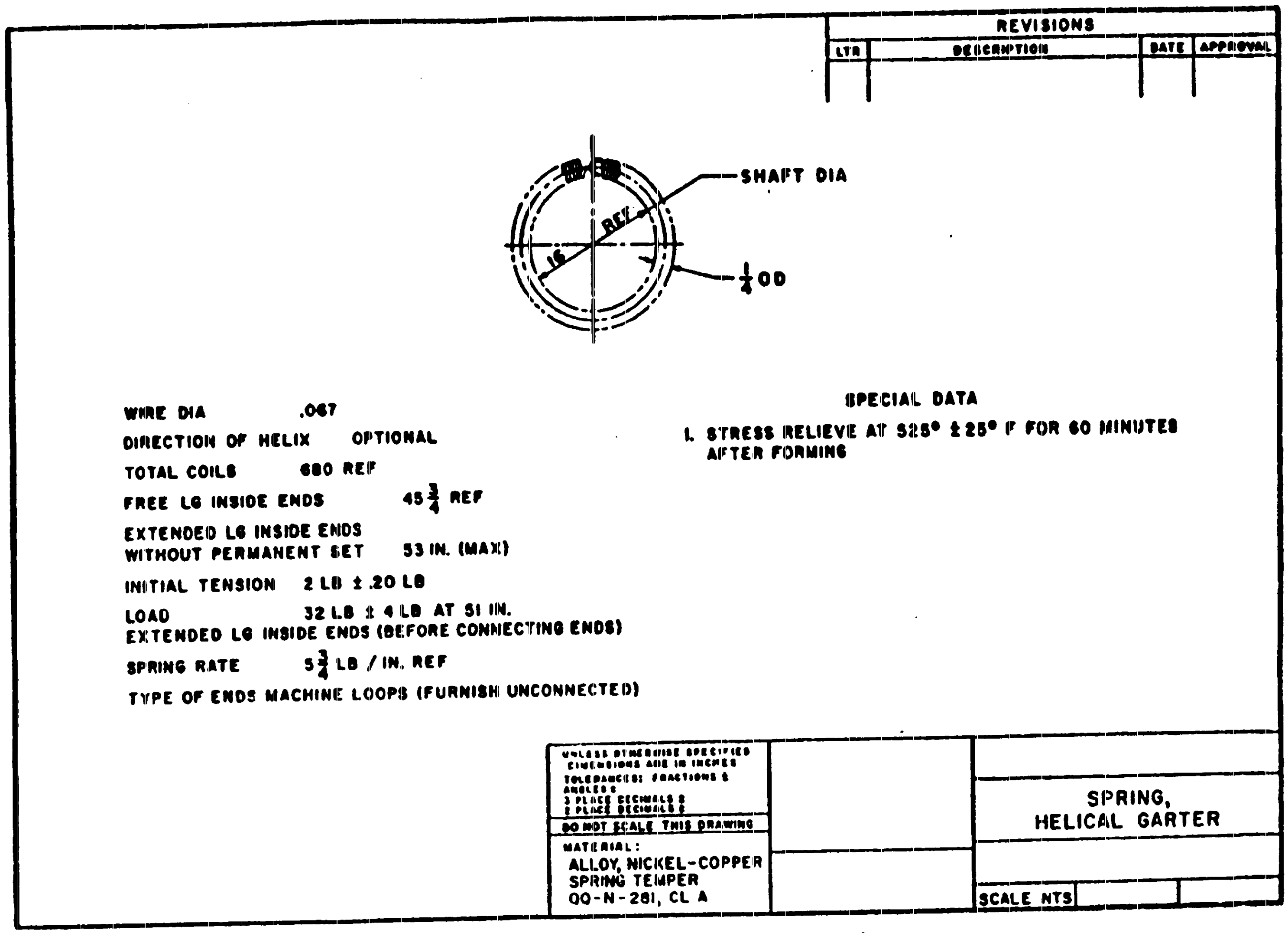


FIGURE 30. *Typical detail drawing of garter spring*

APPENDIX A

THIS APPENDIX IS FOR THE PURPOSE OF PROVIDING GUIDANCE TO DESIGN ENGINEERS AND INCLUDES NO MANDATORY PROVISIONS.

10. DESIGN OF MECHANICAL SPRINGS

10.1 PURPOSE. The purpose of this appendix is to provide ready reference data to spring design information including data on materials, design, manufacturing processes and recommended tolerances. The information contained herein is intended to provide guidance to the designers of springs and not to impose restrictions in design.

10.2 SCOPE. The data contained in this appendix are sufficient for general spring design. This appendix is divided into three sections: Section I — MATERIALS Section II — DESIGN, and Section III — MANUFACTURE.

APPENDIX A, SECTION I

11. SPRING MATERIALS

11.1 GENERAL. More than forty different steel compositions, seven corrosion resisting steels and about twenty different non-ferrous compositions are used as spring materials. Each has some particular advantage over the others and the selection is based on seeking, in addition to other requirements, the best behavior in applications involving high stress, shock loads, elevated temperatures and corrosion resistance. All spring wire should be used in round cross-section whenever possible as it is more readily available, less expensive, usually has higher mechanical properties and is easier to fabricate. Steel wire has the longest fatigue life and can withstand the highest stresses. Copper-base alloys have the best electrical conductivity combined with good corrosion resistance. Corrosion resisting steels have good resistance to moderately elevated temperatures and to corrosion. Nickel-base alloys have the best resistance to elevated temperatures combined with excellent corrosion resistance. More specific data on these materials is covered in this section. The SAE and AISI references and the specifications of bar in the following paragraphs represent typical compositions of metal from which springs with the specified mechanical properties can be produced.

11.2 SIZES AVAILABLE. In the column headed Mechanical Properties in Paragraphs 12.2 through 17.3 are shown "Sizes Available", which lists the range of sizes usually available from warehouses or wire mills. For most materials both smaller and larger sizes can be obtained on special order. The sizes shown in Table I, Appendix A, Section I, should be used wherever possible.

12. HIGH CARBON STEELS

12.1 GENERAL. The most commonly used of all spring materials are the high carbon spring steels. These materials are used in Music-Wire, Hard-Drawn Wire, Oil Tempered Wire and Valve Spring Wire. All are readily available in a wide variety of sizes. These materials should be used in preference to other materials whenever the design requirements permit their use. They are not usually recommended for sub zero temperature applications.

12.2 MUSIC WIRE QQ-W-470, ASTM A228-51

Mechanical Properties:

Modulus;
In tension
$E=30 \times 10^6$
In torsion
G (Up to .100 in.)
$=12 \times 10^6$
G (Over .100 in.)
$=11.75 \times 10^6$

Elastic Limit
Tension = 65 to 75% of TS
Torsion = 45 to 50% of TS

Density = .284 lb/in.^{3}

Maximum elevated temperature
250° F

Sizes available; .004 to .180

Music wire is the highest quality cold drawn, high carbon wire in the ferrous group. It has exceptionally high tensile strength and uniform temper. It is recommended for small springs which are subjected to high stresses and suddenly applied loads. Only round sections have high tensile strengths. This material is not a true music wire if obtained in any sections other than round.

12.3 HARD DRAWN STEEL WIRE QQ-W-428, TYPE II; ASTM A227-47

Mechanical Properties:

Modulus;
In tension
$E=30 \times 10^6$
In torsion
$G=11.5 \times 10^6$

Elastic Limit;
Tension = 60 to 70% of TS
Torsion = 45 to 55% of TS

Density = .284 lb/in.^{3}

Maximum elevated temperature
250° F

Sizes Available: .028 to .625 in. dia

Hard drawn steel wire is a low cost spring steel which can be used for average stress applications where accuracy of wire diameter, spring diameter and loads are not required with great precision. Do not use in applications where long life is required. This material can be readily electroplated. Square sections are also obtainable but at reduced tensile strengths.

12.4 OIL TEMPERED STEEL WIRE QQ-W-428, TYPE I; ASTM A229-56

Mechanical Properties:

Modulus;
In tension
$E=28.5 \times 10^6$
In torsion
$G=11.2 \times 10^6$

Oil-tempered steel wire is a general purpose spring steel of uniform quality and temper. Because of its surfac condition this material is recommended for springs where: (1) the stress requirements are not too extreme, (2) the spring is not subjected to impack or shock loading, and (3) the spring index is 5 or greater. This material is widely used for me-

Elastic Limit;
Tension=80 to 90% of T S
Torsion=40 to 50% of T S

Rockwell Hardness=C45–50 less than .125 in dia
C42–48 .126 in. to .250 in. dia
C40–45 .251 in. to .500 in. dia

Density=.284 lb/in.^{3}

Maximum elevated temperature
850° F

Size Available: .032 to .500 in. dia and 1/32 to 1/2 in square sections in common fractions.

chanical products and machine tools in applications where the slight scale on the surface is not objectionable. The scale can be removed if necessary by shot blasting. Also obtainable in the annealed condition if desired.

12.5 CARBON-STEEL VALVE SPRING QUALITY WIRE ASTM A230–47

Mechanical Properties:

Modulus;
In tension
E=29.5 x 10^6
In torsion
G=11.2 x 10^6

Elastic Limit;
Tension=85 to 95% of T S
Torsion=50 to 60% of T S

Rockwell Hardness=C45–50 for .125 in. diameter and under
C42–48 above .125 in.

Density=.284 lb/in.^{3}

Maximum elevated temperature
850° F

Sizes Available:
.093 to .250 in. dia

Because of the absence of scale on this material, it is suitable for use where flaking of scale would clog or otherwise injure moving parts. This material is recommended for springs of infinite life with an average stress range. This material is available in the tempered and annealed condition, but the tempered condition is most often used. The annealed condition should be used with subsequent hardening and tempering, for springs with a low spring index (less than 5) or where severe forming is required.

13. ALLOY STEEL WIRE AND BAR

13.1 GENERAL. Alloy spring steels are particularly useful in applications involving high stress and where shock or impact loadings occur. They can also withstand higher and lower temperature conditions than the carbon spring steels. All these alloys are generally used in the oil tempered or pre-hardened condition. They also are obtainable in the annealed condition for springs that have sharp bends or small spring indexes such as 5 or less.

13.2 CHROME-VANADIUM ALLOY STEEL WIRE & BAR ASTM A231-41 (Wire); QQ-S-624 FS 6150; A60-49 (Bars); ASTM A232-47 (Wire); QQ-W-412, Comp 1 (Wire)

Mechanical Properties:

Modulus;
In tension
$E=29.5 \times 10^6$
In torsion
$G=11.2 \times 10^6$

Elastic Limit;
Tension=88 to 93% of T S
Torsion=65 to 75% of T S

Rockwell Hardness=C45-50

Density=.284 lb/in.^{3}

Maximum elevated temperature
425° F

Sizes Available:
.032 to .375 in. dia in the tempered or annealed condition; $\frac{7}{16}$ to 2 in. dia in bars for hot rolled springs. Square sections in common fractional sizes are also obtainable.

Chrome-Vanadium alloy steel wire will withstand higher stresses than Oil-tempered steel wire, but not as high as music wire. This material is recommended for springs which are subjected to shock by impact or suddenly applied loads, and for springs which are subjected to a large number of stress cycles. This material is frequently used for die springs. It is especially useful when annealed round wire is flattened into rectangular section with round edges. Die springs made from such sections are hardened and tempered after coiling and then shot-peened to increase their endurance limit. Hot rolled bars in heavy sections are also used for hot rolled springs and torsion bars.

13.3 SILICO-MANGANESE ALLOY STEEL BAR SAE 9260 & ASTM A59-49 (BAR)

Mechanical Properties:

Modulus;
In tension
$E=29.5 \times 10^6$
In torsion
$G=11.2 \times 10^6$

Elastic Limit;
In tension=78 to 86% of T S
In torsion=55 to 65% of T S

Rockwell Hardness=C44 to 48

Density=.284 lb/in.^{3}

Sizes Available:
$\frac{7}{16}$ to 2 in. dia bars

This alloy steel has frequently been used as a less expensive substitute or alternate for chrome-vanadium. It does not have mechanical properties quite as high as other alloy steels. This alloy is obtainable in the oil-tempered, and in the annealed conditions in hot rolled bars. Heavy flat sections have been used for leaf springs and torsion bars.

13.4 CHROME-SILICON ALLOY STEEL WIRE QQ-W-412, COMP 2, TYPE II; ASTM A401-58

Mechanical Properties:

Modulus;
In tension
$E = 29.5 \times 10^6$
In torsion
$G = 11.2 \times 10^6$

Elastic Limit;
In tension = 88 to 93% of T S
In torsion = 65 to 75% of T S

Rockwell Hardness = C50 to 53

Density = .284 lb/in.^{3}

Maximum elevated temperature
475° F

Sizes Available:
.032 to .375 in. dia wire.

This alloy steel is especially suited for highly stressed springs subjected to shock or impact loading such as recoil springs in anti-aircraft guns and for moderately elevated temperatures. It may be obtained in the oil-tempered condition, but has most often been used in the annealed condition and then hardened and tempered to quite high hardnesses after coiling.

14. CORROSION RESISTING STEEL WIRE

14.1 GENERAL. Corrosion resisting steels are especially useful in applications involving corrosion and high temperatures. The 300 series "18-8" chromium-nickel austenitic types are usually used up to about $^{3}/_{16}$ inch diameter and the 400 series chromium martensitic types for larger sizes. The 300 series "18-8" types are hard-drawn to high tensile strengths and cannot be hardened by heat treatment. The 400 series martensitic types are usually used in the annealed condition and then hardened and tempered after coiling. The 300 series are non-magnetic in the fully annealed condition only — hard drawing to obtain spring qualities cause some magnetic ability which cannot be totally removed. The 400 series are magnetic and should not be used at sub zero temperatures. Passivating or immunizing by dipping in a 20 to 40% solution of nitric acid after fabrication is desirable for flat strip, but is frequently omitted on springs made of round wire.

14.2 CORROSION-RESISTING STEEL WIRE QQ-W-423, Composition FS 302, Condition B; AISI 302; SAE 30302; ASTM A313-55

Mechanical Properties:

Modulus;
In tension
$E = 28 \times 10^6$
In torsion
$G = 10 \times 10^6$

Elastic Limit;
In tension = 65 to 75% of T S
In torsion = 45 to 55% of T S

This is a general purpose corrosion resisting material. Its spring properties are developed by cold working only. This material has higher tensile strengths than FS 304 and FS 316 but does not possess quite as good corrosion resisting properties. It is slightly magnetic in the spring temper, has low creep and resists relaxation at elevated temperatures. This material is also available in strip form for flat springs.

Rockwell Hardness = C42–47

Density = .288 lb/in.^{3}

Maximum elevated temperature
550° F

Sizes Available:
.005 to .375 in. dia
Full Hard Temper up to .375 in. dia
Above .375 in. dia at lower hardness

14.3 CORROSION-RESISTING STEEL WIRE QQ–W–423, Composition FS 304, Condition B; AISI 304; SAE 30304

Mechanical Properties:
The same as FS 302, but the tensile strengths are about 5% lower.

This material is quite similar to FS 302, has somewhat lower tensile strength, but has better bending properties and is an excellent alternate for most applications. The two types are often used interchangeably, however, FS 304 has somewhat better corrosion resistance than FS 302.

14.4 CORROSION-RESISTING STEEL WIRE QQ–W–423, Composition FS 316, Condition B; AISI 316; SAE 30316

Mechanical Properties:
The same as FS 302, but the tensile strengths are about 10 to 15% lower.

This material has better corrosion resistance properties and is less magnetic than composition FS 302. Its mechanical properties at sub zero temperatures are better, but it cannot be used at stresses as high as FS 302. It has ability to take sharp bends due to the lower tensile strength.

14.5 CORROSION-RESISTING STEEL WIRE 17–7 PH

Mechanical Properties:

Modulus;
In tension
$E = 29.5 \times 10^6$
In torsion
$G = 11 \times 10^6$

Elastic Limit;
Tension = 75 to 80% of T S
Torsion = 55 to 60% of T S

Rockwell Hardness = C47–50
after hardening

Density = .277 lb/in.^{3}

Sizes Available:
.030 to .375 in. dia

This new type of corrosion resisting steel is of the precipitation hardening type. It is obtainable in the annealed condition, but is most often used in the cold worked condition and then precipitation hardened at 900° F for 1 hour. It then has tensile strengths higher than FS 302 and in some smaller sizes the tensile strength is equal to that of music wire.

14.6 CORROSION-RESISTING STEEL WIRE QQ-W-423, Composition FS 420; AISI 420; SAE 51420

Mechanical Properties:

Modulus;
In tension
$E = 29 \times 10^6$
In torsion
$G = 11.20 \times 10^6$

Elastic Limit;
Tension = 65 to 75% of T S
Torsion = 45 to 55% of T S

Rockwell Hardness = C46–51 after hardening

Density = .280 lb/in.^{3}

Sizes Available:
.030 to .500 in. dia

This corrosion resisting steel is usually used in diameters above 3/16 inch, but occasionally it is used in smaller sizes such as .057 inch for recoil springs in rifles. It is always used in the annealed condition and then hardened and tempered after coiling. This material does not have corrosion resisting properties until after hardening. Clean bright surfaces provide best corrosion resistance and heat treating scale should be removed whenever possible.

14.7 CORROSION-RESISTING STEEL WIRE AISI 431; SAE 51431

Mechanical Properties:

Modulus;
In tension
$E = 30 \times 10^6$
$G = 11.5 \times 10^6$

Elastic Limit;
Tension = 72 to 75% of T S
Torsion = 50 to 55% of T S

Rockwell Hardness = C47–51

Density = .280 lb/in.^{3}

Sizes Available:
.050 to .312 in. dia

This new type corrosion resisting steel is first hardened and tempered and then cold drawn. This combination produces bright clean wire with high tensile strengths nearly as high as music wire, but its corrosion resistance is not quite equal to FS 302. The hardened wire is magnetic and finds many uses for springs subjected to high stresses. Flat strip is also available.

15. COPPER-BASE ALLOYS

15.1 GENERAL. Copper-base alloys combine good electrical properties with excellent corrosion resistance. Although more expensive than steel they find many uses in electrical components and are excellent at sub zero temperature applications. All are non-magnetic.

15.2 SPRING BRASS WIRE QQ-W-321, Composition B, Spring Temper; ASTM B134-52, Alloy 6; SAE 80A

Mechanical Properties:

Modulus;
In tension
$E = 15 \times 10^6$
In torsion
$G = 5.0 \times 10^6$

Elastic Limit;
In tension = 75 to 80% of T S
In torsion = 45 to 50% of T S

Rockwell Hardness = B89–95

Density = .308 lb/in.^{3}

Maximum elevated temperature
150° F

Sizes Available:
.005 to .500 in. dia

Spring brass is recommended for use where high electrical conductivity, excellent corrosion resistance, ease of forming, low cost, and repeated flexure at low stress and low temperatures are required. Spring brass is non-magnetic. It is not recommended for applications where the stress is severe. It is also available in flat strip.

15.3 PHOSPHOR BRONZE WIRE QQ-W-401; ASTM B159-58, ALLOY A; SOE 81

Mechanical Properties:

Modulus;
In tension
$E = 15 \times 10^6$
In torsion
$G = 6.0 \times 10^6$

Elastic Limit;
Tension = 75 to 80% of T S
Torsion = 45 to 50% of T S

Rockwell Hardness = B90–97

Density = .320 lb/in.^{3}

Maximum elevated temperature
212° F

Sizes Available:
.005 to .500 in. dia

Phosphor bronze is used for electrical conductivity, ability to resist corrosion, and non-magnetic properties. It has good bending and forming properties and the ability to withstand repeated flexures. This is the most popular of the copper-base non-ferrous alloys. It is also available in flat strip for electrical contact fingers. Superfine grain structure withstands quite high stresses and has long endurance to repeated flexure

15.4 BERYLLIUM COPPER WIRE QQ-C-530; ASTM B197-52

Mechanical Properties:
Modulus;
In tension
$E = 19 \times 10^6$

Beryllium copper is a precipitation hardening non-magnetic material with good electrical conductivity and corrosion resistance. It is non-magnetic and has high elastic and fatigue strength, and low drift and

In torsion
$G = 7.3 \times 10^6$

Elastic Limit;
Tension = 75% of T S
Torsion = 50 to 55% of T S

Rockwell Hardness = C37 to 41

Density = .298 lb/in.^{3}

Maximum elevated temperature
300° F

Sizes Available:
.005 to .500 in. dia

hysteresis. This material can be severely formed in the annealed condition. Following forming, beryllium copper is precipitation hardened at 600° F for 2 hours. Pretempered wire is also useful for many applications where the stresses are not too high. Flat strip is also available.

16. NICKEL-BASE ALLOYS

16.1 GENERAL. Nickel-base alloys have excellent corrosion resistance combined with ability to withstand both elevated and below-zero temperature applications. Also, their non-magnetic characteristic is important for such devices as gyroscopes, chronoscopes and indicating instruments. These materials have high electrical resistance and should not be used for conductors of electrical current.

16.2 NICKEL-COPPER ALLOY, QQ-N-281, CLASS A, SPRING TEMPER

Mechanical Properties:

Modulus;
In tension
$E = 26.0 \times 10^6$
In torsion
$G = 9.5 \times 10^6$

Elastic Limit;
Tension = 65 to 70% of T S
Torsio = 38 to 42% of T S

T S = 165,000 up to .028 in. dia incl
160,000 over .028 in. to .057 in. dia incl
150,000 over .057 in. to .114 in. dia incl
140,000 over .114 in. to .312 in. dia incl
135,000 over .312 in. to .375 in. dia incl
130,000 over .375 in. to .500 in. dia incl
120,000 over .500 in. to .563 in. dial inc

Density = .319 lb/in.^{3}
Maximum elevated temperature
400° F (450° F for short periods)

Sizes Available:
.005 to .250 in. dia full hard, over .250 at lower hardness.

Monel is recommended for use at elevated temperatures where corrosive conditions exist. It retains its corrosive resistance properties at relatively high temperatures. This material has good resistance to sea water attack. It is a hard drawn wire and cannot be heat treated for hardening. Flat strip is also available. It is nearly non-magnetic.

16.3 NICKEL-COPPER-ALUMINUM ALLOY, WROUGHT, QQ-N-286, CLASS A

Mechanical Properties:

Modulus;
In tension
$E = 26.0 \times 10^6$
In torsion
$G = 9.5 \times 10^6$

Elastic Limit;
Tension = 65 to 70% of T S
Torsion = 38 to 42% of T S

S = 180,000 up to .114 in dia incl
170,000 over .114 in. to .375 in. dia incl
160,000 over 375 in. to .563 in. dia incl

Density = .305 lb/in.^{3}

Maximum elevated temperature
450° F. (500° F. for short periods)

Sizes Available:
.005 to .563 in. dia for cold wound springs
⅝ in. dia and over for hot wound springs

K-Monel is a precipitation hardening alloy with excellent resistance to both corrosion and moderately elevated temperatures. It is usually used in larger sizes than Monel, as it can be formed in the cold drawn condition and then heat treated for hardening purposes. The mechanical properties are for the spring temper, age hardened condition. This alloy is non-magnetic.

16.4 WIRE: NICKEL-CHROMIUM-IRON ALLOY, QQ-W-390, CONDITION C

Mechanical Properties:

Modulus;
In tension
$E = 31 \times 10^6$ at 70° F.
29.5×10^6 at 400° F.
29.2×10^6 at 500° F.
28.7×10^6 at 600° F.
28.2×10^6 at 700° F.
28.0×10^6 at 750° F.

In torsion
$G = 11.2 \times 10^6$ at 70° F.
10.8×10^6 at 400° F.
10.5×10^6 at 500° F.
10.2×10^6 at 600° F.
10.0×10^6 at 700° F.
9.8×10^6 at 750° F.

Elastic Limit;
In tension = 65 to 70% of T S
In torsion = 40 to 45% of T S

T S = 185,000 up to .057 in dia incl
175,000 over .057 in. to .114 in dia incl
170,000 over .114 in. to .229 in. dia incl

Inconel is recommended for springs requiring: (1) the retention of spring properties at relatively high temperatures, (2) corrosion resistance, (3) low magnetic permeability. This is one of the most popular of the nickel-base alloy group. It is cold drawn and cannot be hardened by heat treatment. Wire diameters up to ¼ inch are most often used. This alloy is non-magnetic.

165,000 over .229 in. to .329 in. dia incl
160,000 over .329 in. to .375 in. dia incl
155,000 over .375 in. to .500 in dia incl
140,000 over .500 in. to .563 in. dia incl

Density = .304 lb/in.^{3}

Maximum elevated temperature
650° F. (750° F. for short periods)

Sizes Available:
.010 to .1875 in. dia full hard, over .1875 at lower hardness.

16.5 WIRE: NICKEL-ALLOY, SPRING, AMS-5698 for No. 1 temper; AMS-5699 for spring temper

Mechanical Properties:

Modulus;

In tension

E = 31.00 x 10^6 at 70° F.
28.1 x 10^6 at 700° F.
27.7 x 10^6 at 800° F.
27.2 x 10^6 at 900° F.
26.7 x 10^6 at 1000° F.
26.1 x 10^6 at 1100° F.
25.5 x 10^6 at 1200° F.

In torsion
G = 11.2 x 10^6 at 70° F.
10.0 x 10^6 at 700° F.
9.8 x 10^6 at 800° F.
9.4 x 10^6 at 900° F.
9.1 x 10^6 at 1000° F.
8.8 x 10^6 at 1100° F.

Elastic Limit;
Tension = 65 to 70% of T S
Torsion = 40 to 45% of T S

T S = For No. 1 temper:
155,000 min up to .025 in. dia incl
165,000 min over .025 to .468 in. dia
For spring temper:
220,000 min for .012 to .250 in. dia incl
200,000 min over .250 to .418 in. dia incl

Density = .298 lb/in.^{3}

Maximum elevated temperature
1150 to 1200° F at low stresses

Inconel-X is a precipitation hardening material with high corrosion resistance and oxydization resistance at elevated temperatures. To make optimum use of this material, it is recommended that it be used in applications where the temperature is from 650 to 1150° F. For applications below 650° F the use of other materials is recommended. Neither cold setting nor shot peening should be specified on product drawing, when springs are intended to be operating within the recommended temperatures (650 to 1150° F) due to the adverse effect of pre-stressing on initial relaxation and the rate of relaxation under heat and stress. Large diameter bars can be hot coiled, and this is often done for 5/8 in dia and larger sizes depending upon the spring index.
This non-magnetic alloy can also be used at sub-zero temperatures. The mechanical properties are for the tempers indicated after age hardening. For sub zero to 700° F applications, use spring temper stock and heat 1200° F for 4 hours, after coiling. For 700 to 1000° F applications, use No. 1 temper (or hot finished stock) and heat 1350° F for 16 hours, after coiling. For 1000 to 1200° F applications, use spring temper stock and heat 2100° F for 2 hours, air cool, then 1550° F for 24 hours, air cool and 1300° F for 20 hours and air cool, after coiling.

Sizes Available:
.012 to .468 in. dia incl No. 1 temper

.012 to .418 in. dia incl spring temper
.375 in. dia and over, hot finished (for hot wound springs)

17. STEEL STRIP, HIGH CARBON

17.1 GENERAL. Several types of flat cold rolled steel strip with different ranges of carbon content, tempers, finishes and edges are obtainable, but only two types are readily available. These two types are used for over 95 per cent of all applications requiring steel strip. Thin sections of high hardness (over Rockwell "C" 43) are highly susceptible to hydrogen embrittlement resulting from electroplating or pickling operations. These materials are used for mainsprings in clocks and timing devices; for constant force springs, clips and stampings. For general information see Spec. MIL-S-17919.

17.2 STEEL STRIP, HIGH CARBON SAE 1074; MIL-S-17919 MATL. No. 1 & 4

Mechanical Properties:
Modulus;
In tension
$E=30 \times 10^6$
In torsion
$G=11.5 \times 10^6$
T S=244,000 to 305,000 for sizes under .032 in.
214,000 to 240,000 for sizes over .032 in.
Elastic Limit;
Tension=85 to 90% of T S
Torsion=65 to 75% of T S
Rockwell Hardness=C46 to 49
Density=.284 lb/in.^{3}
Maximum Elevated Temperature 350° F
Sizes Available:
.005 to .062 in. thickness, others on special order.

This type of material is usually called "cold rolled blue tempered spring steel". It is slit to desired widths and the edges can be squared or rounded as desired. The material obtained in the hardened conditions is widely used for flat springs, spiral, clock and motor springs. It is also available in the annealed condition for use in automatic equipment and where severe forming operations are required, after which it is hardened and tempered.

17.3 STEEL STRIP, HIGH CARBON SAE 1095; MIL-S-17919 MATL. No. 1 & 2

Mechanical Properties:

The same as SAE 1074 but the tensile strengths are about 5 per cent higher.

Rockwell Hardness=C48 to 51

This type of material, usually called "cold rolled blue tempered and polished clock spring steel', is widely used in clocks and motor springs because it can withstand higher stresses than SAE 1074. It is slit to widths desired and the edges are usually made square or rounded. It is generally obtained in the hardened condition, but annealed material can be obtained.

TABLE I. *Preferred sizes for spring materials wire diameters, strip thickness and bars*

Spring steels			Corrosion resisting		Copper and nickel alloys	
Music wire	High carbon & alloy steels	Valve spring quality steels	"18-8" chrome nickel austenitic 300 series	Straight chrome martensitic 400 series	Spring quality brass phosphor bronze beryllium cop. monel & inconel	K monel & inconel X
.004	.032	.092	.004	Same as high carbon & alloy steels, Col. 2	.010	.125
.006	.035	.105	.006		.012	.156
.008	.041	.125	.008		.014	.162
.010	.047	.135	.010		.016	.188
.012	.054	.148	.012		.018	.250
.014	.063	.156	.014		.020	.313
.016	.072	.162	.020		.025	.375
.018	.080	.177	.026		.032	.475
.020	.092	.188	.032		.036	.500
.022	.105	.192	.042		.040	.563
.024	.125	.207	.048		.045	.688
.026	.135	.218	.054		.051	.750
.028	.148	.225	.063		.057	.875
.032	.156	.244	.072		.064	1.000
.042	.162	.250	.080		.072	1.125
.048	.177		.092		.081	1.250
.063	.188		.105		.091	1.375
.072	.192		.120		.102	1.500
.080	.207		.125		.114	1.625
.090	.218		.135		.125	1.750
.107	.225		.148		.128	2.000
.130	.244		.156		.144	
.162	.250		.162		.156	
.177	.263		.177		.162	
	.283		.188		.182	
	.307		.192		.188	
	.313		.207		.250	
	.362		.218			
	.375		.225			
			.250			
			.312			
			.375			

Note 1. Square sizes of wire and bars are obtainable in preferred fractional sizes as follows: 1/32, 1/16, 3/32, 1/8, 5/32, 3/16, 7/32, 1/4, 9/32, 5/16, 3/8, 7/16, 1/2, 9/16, 5/8, 11/16, 3/4, 13/16, 7/8, 15/16, 1, 1 1/16, 1 1/8, 1 3/16, 1 1/4, 1 5/16, 1 3/8, 1 1/2, 1 5/8, 1 3/4, 1 7/8 and 2 in.

Note 2. Music Wire and Valve spring quality wire are available in round sections only.

Note 3. Music Wire is drawn to American Steel Wire & Music Wire Gage and to special gages. Steel wire is drawn to U.S. Steel Wire Gage (same as Washburn & Moen Gage). Copper and Nickel Alloys are drawin to the American Wire Gage (same as Brown & Sharpe Gage).

APPENDIX A, SECTION II

20. SPRING DESIGN

20.1 GENERAL. The proper design of springs requires an understanding of 1 — spring materials, 2 — design formulas and stress analysis and 3 — manufacture. Various aids to designers are available including special spring slide rules, tables of constants, curves, charts and nomographs. All are helpful, but an understanding of the basic fundamental formulas and experience in their use is essential to good design. Except for a few sizes of valve and die springs, there are very few springs manufactured for stock because of the infinite variety of characteristics involved. Utmost care in their design and manufacture and thorough analysis of service conditions are required for satisfactory performance.

20.1.1 Purpose. The purpose of this section is to describe the design methods used for each type of spring commonly used.

20.1.2 Scope. The data in this appendix are sufficient for general design purposes, and is not intended to include information for unusual designs or seldom used types of springs.

201.3 Abbreviations and Symbols. The following abbreviations and symbols are used throughout the appendix, unless otherwise noted:

A	= constant, for rectangular wire.
B	= constant, for rectangular wire.
b	= breadth or width, in.
C	= Spring index = D/d.
CL	= compressed length, in.
D	= mean coil diameter, in.
d	= diameter of wire or side of square, in.
E	= modulus of elasticity in tension, psi
F	= deflection, for N coils with load P, in.
F°	= deflection, for N coils, rotary, deg.
FL	= free length, unloaded spring, in.
f	= deflection, for one active coil, in., at load P.
G	= modulus of elasticity in torsion, psi.
ID	= inside diameter, in.
in.	= inch.
K	= curvature stress-correction factor
L	= active length subject to deflection, in.
l	= length, in.
lb	= pound
M	= bending moment ,in. lb.
N	= total active coils
n'	= vibration per minute.
OD	= outside diameter, in.
P	= load, lb.
P_1	= applied load, lb (also P_2, etc.).
p	= pitch, in.
psi	= pounds per square inch.
R	= distance from load to central axis, in.

r	= spring rate, load per inch, lb/in.
r_t	= spring rate, inch lbs per deg. (Torsion springs).
S_b	= stress, bending, psi.
S_t	= stress, torsional psi.
S_{it}	= stress, torsional, due to initial tension, psi.
SG	= squared and ground.
SH	= solid height, in. (or SL = solid length).
s	= height, load is dropped, in.
T	= torque = P x R, lb. in.
TC	= total coils.
t	= thickness, in.
U	= number of revolutions = F° ÷ 360°.
W	= weight, lb (also applied dynamic load).
x	= multiplied by
Y	= constant, for coned disc (Belleville) springs.
Z_1	= constant, for coned disc (Belleville) springs.
Z_2	= constant, for coned disc (Belleville) springs.
α	= alpha, angle of movement, deg.
π	= pi, 3.1416, in.
σ	= sigma, Poissons Ratio, 0.3 for steel.

21. COMPRESSION SPRINGS

21.1 DESIGN FORMULAS. The design formulas in Table II are used in the design of helical compression and extention springs. Note that the same formulas apply to both types of springs. The formulas in Table III are for determining compression spring dimensions only.

21.2 Compression Spring Ends. Figure 3 illustrates the types of ends on compression springs. Their characteristics follow:

(a) **Open Ends Not Ground**; also called Plain Ends, has the largest eccentricity of loading. These are used only when accuracy of loads is not important. This type is seldom used because such springs tangle severely during shipping.

(b) **Closed Ends Not Ground**; also called squared ends, cost approximately the same as open end type and have less eccentricity. This type is often used on light wire springs under 1/32 in. dia wire and for heavier wire where the index exceeds 13.

(c) **Open Ends Ground**; also called Plain Ends Ground, are seldom used as they cost about the same as the closed ends ground, but have high eccentricity of loading and tangle during shipping. They are sometimes used where the solid height is very limited and it is necessary to have as many active coils as possible in the least space.

(d) **Closed Ends Ground**; also called Squared and Ground, is the most popular type as it provides a level seat and reduces the tendency to buckle. This is the most expensive type and should be avoided for springs made from very light wire. Each end coil is ground for 270° plus or minus 30°.

21.3 DIAMETER CHANGES IN COMPRESSION SPRINGS. When a helical compression spring is compressed an increase in the outside diameter occurs because the angularity of the coils changes so that it is nearly at a right angle to the axis. The outside diameter, when the spring is compressed solid, can be obtained from the following formula:

$$OD_s = \sqrt{D^2 + \frac{p^2 - d^2}{\pi^2}} + d$$

In which:

OD_s = outside diameter at solid length

TABLE II. *Formulas for compression springs and extension springs without initial tension*

Property	Round wire	Square wire	Rectangular wire *
Torsional stress, psi	$\frac{P D}{0.393 d^3}$	$\frac{P D}{0.416 d^3}$	$\frac{P D}{B b t^2}$
S_t	$\frac{G d F}{\pi N D^2}$	$\frac{G d F}{2.32 N D^2}$	$\frac{A G t F}{N D^2}$
Deflection, in.	$\frac{8 P N D^3}{G d^4}$	$\frac{5.58 P N D^3}{G d^4}$	$\frac{S_t N D^2}{A G t}$
F	$\frac{\pi S_t N D^2}{G d}$	$\frac{2.32 S_t N D^2}{G d}$	....
Change in load lb $P_2 - P_1$ Extension springs only	$\frac{\frac{L_1 - L_2}{F}}{P}$	$\frac{\frac{L_1 - L_2}{F}}{P}$	$\frac{\frac{L_1 - L_2}{F}}{P}$
Change in load lb $P_2 - P_1$ Extension springs only	$\frac{\frac{L_2 - L_1}{F}}{P}$	$\frac{\frac{L_2 - L_1}{F}}{P}$	$\frac{\frac{L_2 - L_1}{F}}{P}$
Stress due to initial tension, psi S_{it}	$\frac{S_t}{P} \times IT$	$\frac{S_t}{P} \times IT$	$\frac{S_t}{P} \times IT$
Rate lb/in. r	$\frac{P}{F}$	$\frac{P}{F}$	$\frac{P}{F}$

* See figure 36.

TABLE III. *Compression spring formulas for dimensional characteristics*

Dimensional characteristics	Type of ends			
	Open or plain (not ground)	Open or plain with ends ground	Square or closed (not ground)	Closed and ground
Pitch (p)	$\frac{FL - d}{N}$	$\frac{FL}{TC}$	$\frac{FL - 3d}{N}$	$\frac{FL - 2d}{N}$
Solid Height (SH)	$(TC + 1)\ d$	$TC \times d$	$(TC + 1)\ d$	$TC \times d$
Active Coils (N)	$N = TC$ or $\frac{FL - d}{p}$	$N = TC - 1$ or $\frac{FL}{p} - 1$	$N = TC - 2$ or $\frac{FL - 3d}{p}$	$N = TC - 2$ or $\frac{FL - 2d}{p}$
Total Coils (TC)	$\frac{FL - d}{p}$	$\frac{FL}{p}$	$\frac{FL - 3d}{p} + 2$	$\frac{FL - 2d}{p} + 2$
Free Length (FL)	$(p \times TC) + d$	$p \times TC$	$(p \times N) + 3d$	$(p \times N) + 2d$

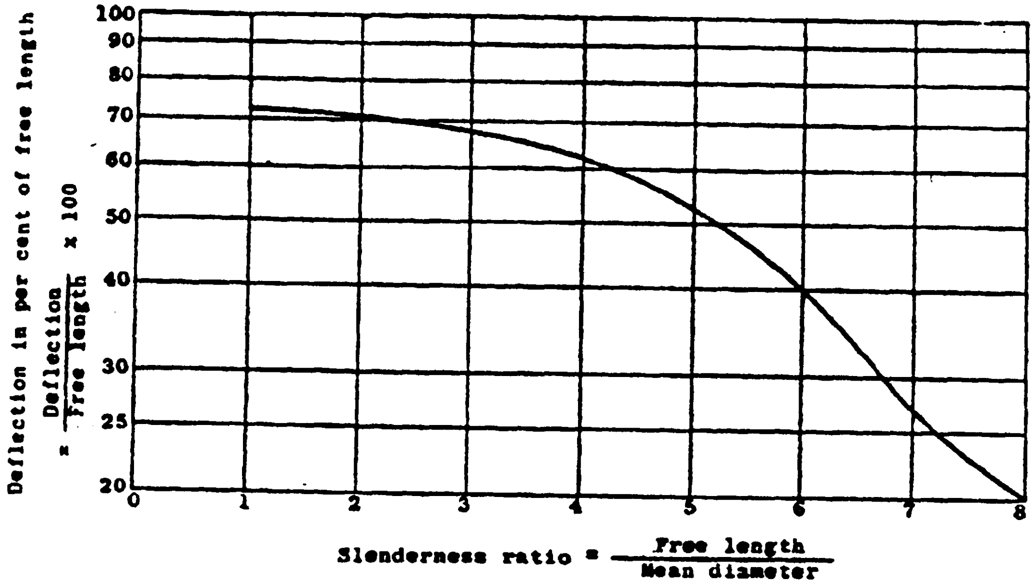


FIGURE 31

p = pitch, at free length
D = mean coil diameter at free length
d = wire diameter

21.4 BUCKLING. Compression springs having a free length greater than four (4) times their mean diameter become critical in lateral stability. When deflected beyond a certain percentage of the free length a spring will buckle. Figure 31 shows the maximum deflection which may be expected without buckling if the ends of the spring are closed and ground. Buckling can be reduced, space permitting, by a redesign using a heavier size wire and increasing the diameter of the coil. Buckling causes an undesirable reduction of the load and may cause early spring failure. If properly guided in a cylinder or over a rod, buckling can be reduced, although friction against the guiding member will affect the load and shorten the spring life.

21.5 DIRECTION OF HELIX. Unless functional requirements dictate a definite direction, the helix of compression and extension springs should be specified as optional. To prevent intermeshing of coils when springs operate one inside the other (see Figure 33), the helixes should be specified as opposite hand. For the same reason, springs which operate to slide freely over screw threads should have the helix specified opposite to that of the screw threads, but when a spring screws onto the threads of a screw or bolt, it should have the same helix as that of the screw or bolt.

21.6 NATURAL FREQUENCY, VIBRATION AND SURGE. The use of springs for loads which are applied dynamically, i.e., with impact or rapidly repeated will be in error if the spring is designed on the basis of static or slow loading. The load, stress, deflection, etc., will have been calculated for applications where the load is applied and held, or the rate of load application is below the natural frequency of the spring. Because of the inertia effect of the coils in instances where the load is suddenly applied, the load on the spring does not have time to distribute itself uniformly throughout the mass of the spring. This non-uniform loading causes deflection or a surge wave (see Figure 32) in a few coils of the spring which results in a high stress in this area and a lower stress in the remainder of the spring. In applications of high rate of repeated loading, non-uniform load distribution occurs in the same manner as suddenly applied loads and the natural frequency of vibration of the spring may be excited. The excitation of the natural frequency of vibration, in some instances, may be of such magnitude as to cause the spring coils to clash causing the spring to destroy its constraint on the mechanism. This is known as spring surge.

The following methods may be employed to prevent spring surging:

1. Stiffen spring
   - (a) Increase diameter of wire
   - (b) Decrease mean diameter of spring
   - (c) Decrease number of coils
   - (d) Use square or rectangular wire
2. Use spring nests
3. Use conical spring
4. Reduce or vary the pitch of the coils near the end of the spring
5. Use stranded wire springs

Formulas for natural frequency of steel springs follow:

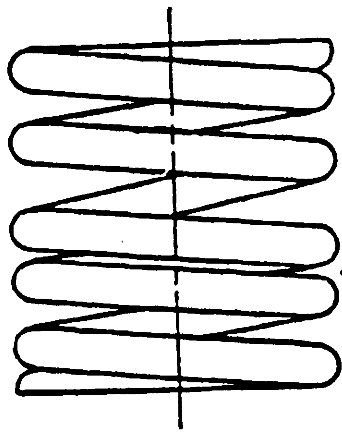


FIGURE 32

UNLOADED SPRING

$$n' = \frac{761{,}500\, d}{N D^2}$$

LOADED SPRING

$$n' = 187.6\sqrt{\frac{1}{F}}$$

(Neglecting spring weight)

If the frequency of the spring and its harmonics are too low, the spring will surge causing the coils to clash. In general, if the natural frequency of the spring is at least thirteen times that of the maximum frequency of the applied load, the design should be satisfactory.

21.7 IMPACT

21.8 SPRING NESTS. The nesting (one in-

TABLE IV. *Formulas for load deflections of compression and extension springs*

Slowly applied load	Suddenly applied load
$F = \frac{W}{r}$	$F = \frac{2W}{r}$
Applied load dropped vertically (spring initially compressed)*	Applied load with striking velocity of V in/sec. (spring in horizontal plane)
$F = \frac{W - P_1 + \sqrt{(W - P_1)^2 + 2Wrs}}{r}$	$F = \frac{-P_1 + \sqrt{P_1^2 + \frac{W r V^2}{386}}}{r}$

* If spring is not initially compressed, disregard P_1

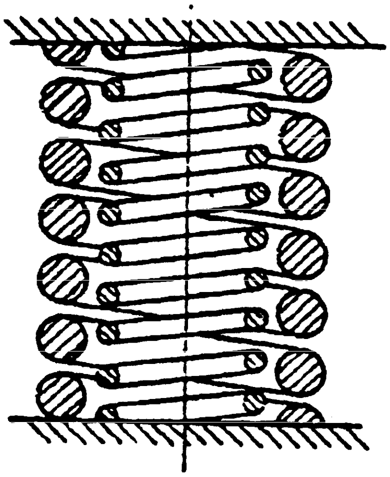


FIGURE 33

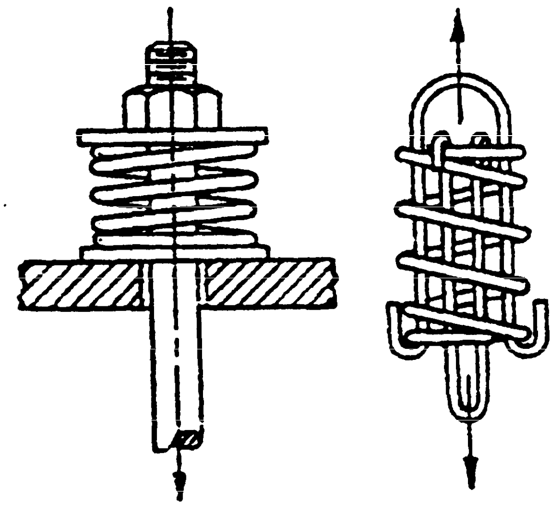


FIGURE 35

side the other) of helical compression springs (see Figure 33) is a method of obtaining maximum energy storage in a limited space. It is desirable to design the springs for equal life with 60 to 70 percent of the load on the outer spring. Maximum energy storage is obtained in a spring nest when the value of the spring indexes are between 5 and 7, when solid lengths of all the springs are approximately the same and when the working stroke ($L_2 - L_1$) is of constant magnitude. The percentage of stress reduction obtained with nested springs is directly proportional to the spring index of the single spring when considering a constant working stroke. Not only is a reduction in the final stress obtained, but the use of nested springs effects a similar reduction in the stress range. Figure 34 shows the variation of the percentage reduction in the final stress and stress range with respect to the spring index of the single spring. The graph is based on the conditions that the nested springs and the single spring have the same values for:

(a)	Active solid height	H_s
(b)	Load-deflection rate	R
(c)	Final Load	P_2
(d)	Modulus of torsion	G
(e)	OD	D_o

The OD of single spring equals the OD of outer spring in nested design.

21.9 COMPRESSION SPRING USED AS AN EXTENSION SPRING. Occassionally certain applications require the action of an extension spring; but the use thereof would produce excessive deflection. This deflection would result in serious distortion, or set,

TABLE V. *Curvature stress correction factors (K) for compression and extension springs*

Spring index D/d	K	Spring index D/d	K	Spring index D/d	K	Spring index D/d	K
3.0	1.580	4.2	1.381	5.8	1.262	7.8	1.189
3.2	1.533	4.4	1.360	6.0	1.252	8.0	1.185
3.4	1.493	4.6	1.342	6.4	1.235	9.0	1.162
3.6	1.459	4.8	1.325	6.8	1.220	10.0	1.145
3.8	1.430	5.0	1.310	7.0	1.213	11.0	1.131
4.0	1.404	5.4	1.284	7.4	1.200	12.0	1.119

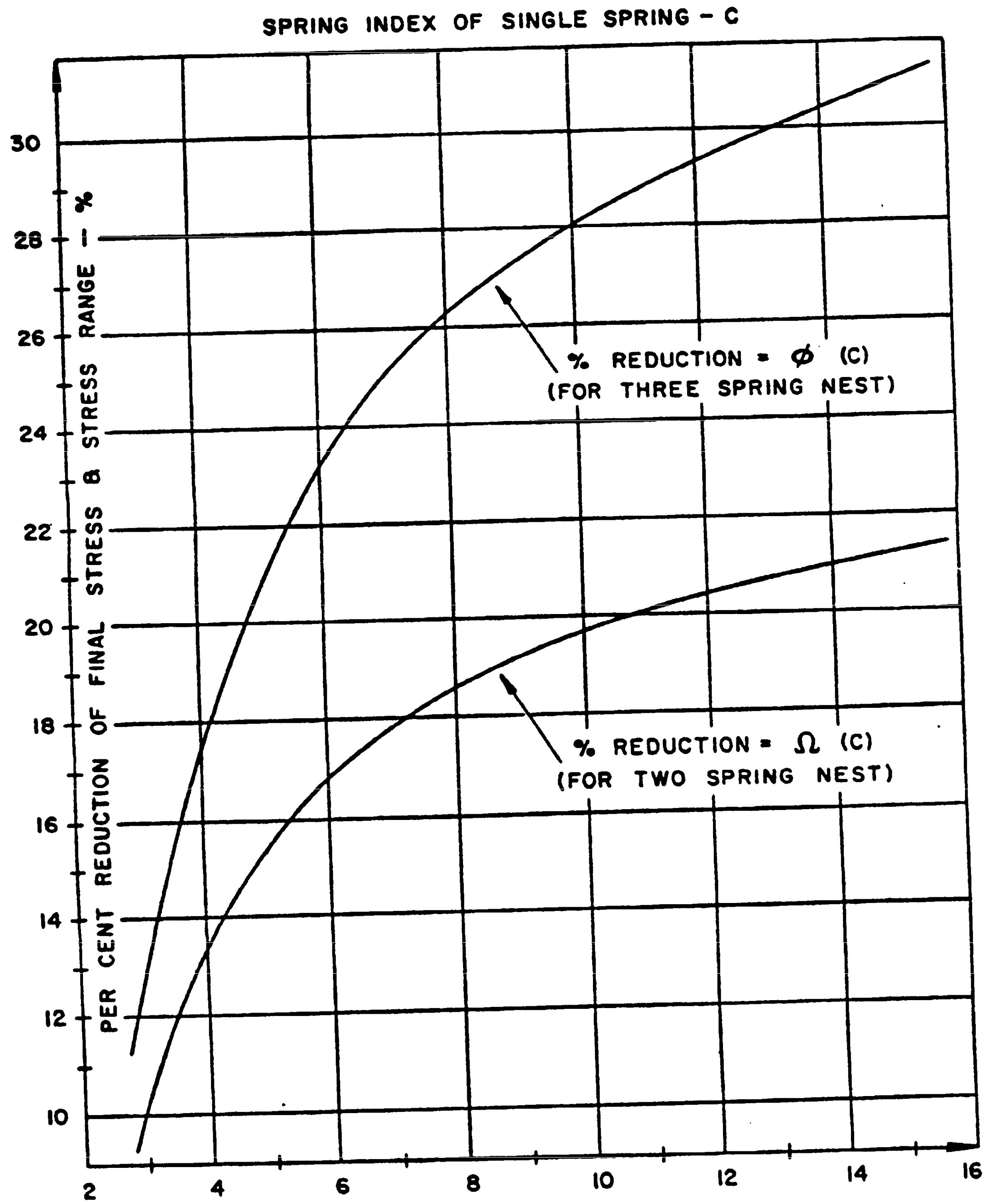


FIGURE 34. *Stress Reduction in spring nests*

of the extension spring and impairment of its fastenings (hooks). Under such circumstances, a compression spring can be used to produce the same action (See Figure 35). Such a device as a through bolt and washer, or a yoke-like drawbar can be utilized, resulting in a spring mount that has the carrying capacity and safety (by virtue of its definite solid length) of a compression spring.

21.10 SPRING INDEX (D/d). The spring index is the ratio of the mean coil diameter of a spring to the wire diameter (D/d). This ratio is one of the most important considerations in spring design inasmuch as the deflection, stress, number of coils, and selection of either annealed or tempered material depends to a considerable extent upon this ratio. The best proportioned springs have an index of 7 to 9. Ratios of 4 to 7 and 9 to 16 are often used. Springs with values larger than 16 require more than standard tolerances for manufacturing; those with values less than 5 are difficult to coil on automatic coiling machines.

21.11 CURVATURE STRESS-CORRECTION FACTORS.

(a) For helical compression and extension springs the curvature stress-correction factor (K) is determined from the following formula:

$$K = \frac{4C - 1}{4C - 4} + \frac{.615}{C}$$

The total stress,
$S_{MAX} = S_t \times K$

(b) For helical torsion springs the curvature stress-correction factor, (K_1), is determined from the following formula:

$$K_1 = \frac{4C^2 - C - 1}{4C\,(C - 1)}$$

The total stress,
$S_{MAX} = S_b \times K_1$

Values of (K) are obtained from Table V and Figure 39. Values of (K_1) are obtained from Figure 51.

EXAMPLE. If a spring with an index of 7.4 has a torsional stress S_t of 80,000 psi, what is the total stress in the spring? From the table it will be found that K equals 1.200; therefore, the total stress equals 80,000 times 1.200 = 96,000 psi. This is the stress that should be compared with allowable stresses to determine whether or no tthe spring is safely designed, and is the sole use made of such data.

In designing a spring it should be borne in mind that the total stress, as determined by this method, should not be used in calculating the deflection or number of coils. The torsional stress $S_t = PD/0.393\ d^3$ or $S_t = G\ d\ F/\pi\ N\ D^2$ should be utilized for such purposes.

21.12 KEYSTONE EFFECT. When square wire and rectangular wire are coiled into springs, a change in shape occurs. This change takes place because some of the material on the outside diameter is drawn into the spring and the material on the inside diameter upsets, thereby changing the wire into a trapezoidal section. The original thickness of the wire is maintained at or near the mean diameter of the coil. It is necessary to take into account this upsetting of the material in determining the solid height of the spring. This dimensional change depends upon the spring index and the thickness of the material and may be determined by the following formula:

$$t' = 0.48t \left(\frac{OD}{D} + 1 \right)$$

t' = new thickness of inner edge after coiling

t = thickness before coiling
This formula may be used for both square and rectangular wire.

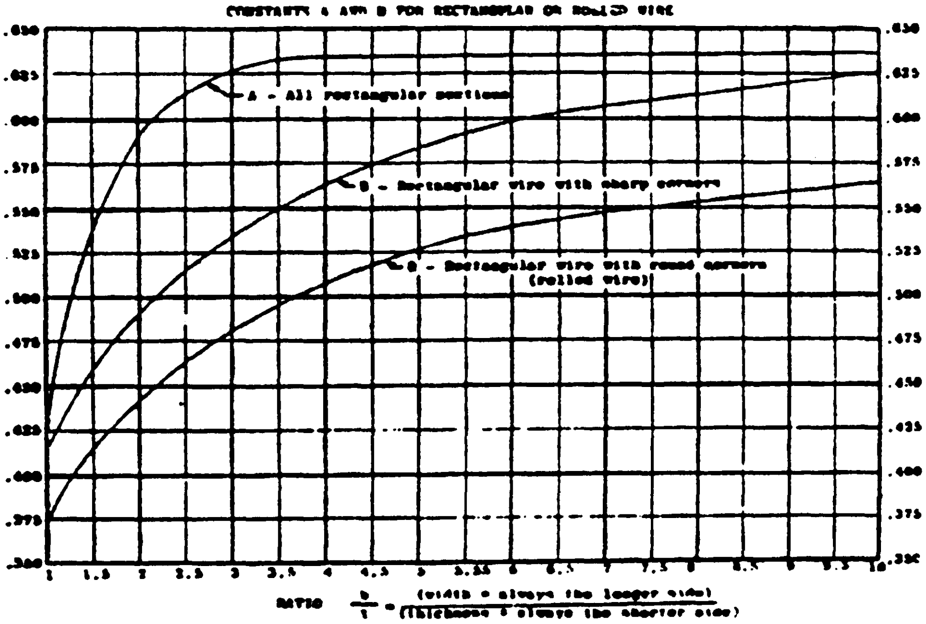


FIGURE 36

21.13 CONSTANTS FOR RECTANGULAR WIRE. The constants A and B in the formulas in Table II for compression and extension springs made from rectangular sections having either sharp or rounded edges are shown in Figure 36.

21.14 PRECAUTIONS AND SUGGESTIONS FOR EFFECTIVE DESIGN OF COMPRESSION SPRINGS.

(a) Compression springs ordinarily should not be permitted to go solid; exceptions occur when they are used as bumpers.

(b) Whenever practicable, springs should be designed so that if they were compressed to the solid length the corrected stress still would not exceed the minimum elastic limit.

(c) The length of a compression spring at maximum working deflection must not be too close to the solid length. As a minimum, a clearance of 10% of the wire diameter should exist between the coils.

(d) The selection of springs for continuous cycling should be made so that the stress range $\frac{(\text{MAX STRESS} - \text{MIN STRESS})}{\text{MAX STRESS}}$ will be as small as possible consistent with other design requirements.

(e) The outside diameter of a compression spring when compressed solid must be less than the minimum hole diameter, if the spring operates in a hole. When operating over a guide the minimum inside diameter must be larger than the maximum diameter of the guide.

(f) The possibility of buckling should be investigated and guides used if necessary.

(g) Use compression springs in preference to other types as they are easier to produce, less expensive and have a deflection limiting feature in the solid length.

(h) The best proportioned springs from the standpoint of manufacture and design have a spring index between 7 and 9, although indexes of 5 to 16 are commonly used.

(i) For indexes less than 5 in the larger diameter wires, it may be necessary to use annealed material and harden after forming.

(j) Specify baking immediately after plating to relieve hydrogen embrittlement.

(k) Three compression springs of identical characteristics standing side by side (in parallel) will have a spring rate and a solid load three times that of one spring.

(l) Three compression springs of identical characteristics placed one on top of another (in series) will have a spring rate only one-third that of one spring and the solid load will be the same as for one spring.

21.15 TABLES OF SPRING CHARACTERISTICS FOR COMPRESSION AND EXTENSION SPRINGS. Table VI, in 5 parts, may be used in the design of helical compression and extension springs made from round wire. The data in Table VI also may be used for square wire by multiplying the deflection per coil f by .707 and the load P by 1.2. In these tables, the upper figure in each box is the deflection f in inches of one coil under a load P in pounds, which is the lower figure in the box. Both terms are based on a torsional modulus G of 11,200,000 psi and on an uncorrected torsional stress of 100,000 psi, which simplifies changing to other stress values.

21.15.1 Example. If a helical compression spring has an OD of 13/16 in. and is made of .041 in. dia wire, the tables shows that at a torsional stress of 100,000 psi such a spring would exert a load of 3.51 lbs and each coil would deflect .407 in. If the spring had 5 active coils, the load would be the same, but the deflection would equal $5 \times .407 = 2.035$ in.

If the allowable stress were only 60,000 psi, both values 3.51 and .407 should be multiplied by .60.

21.16 DESIGN NOMOGRAPHS FOR COMPRESSION AND EXTENSION SPRINGS. As an alternative to Table VI, the nomographs, Figures 37 to 42 may be used in the design of compression and extension springs made from round wire. Note, however, that the values of the torsional modulus G used in Table VI is 11,200,000 psi and the value used in the nomographs is 11,500,000 psi. The design conditions will indicate the type of material required for the application. It will be noted from the formulas for deflection (F) in Table II that the deflection varies inversely as the first power of G. Therefore, for materials having values of G differing from 11.2×10^6 or 11.5×10^6 the value of F (or f) determined from the use of either Table VI or the nomographs must be corrected by multiplying by the ratio of 11.2×10^6 (or 11.5×10^6) to the proper value of G for the material.

21.16.1 Example. Design a spring to develop a load (P_2) at final assembled length (L_2), the OD of the spring being limited to a maximum permissible value and the initial assembled length (L_1) also being known. Thus the approximate mean spring diameter (D) is known. Assume a stress valve somewhat lower than the recommended maximum working stress for the selected spring material and the intended service (deflection cycles). On the nomograph of Figure 37 or 38, connect with a straight line the value of P_2 on the (P) scale with the value of the mean spring diameter on the (D) scale. Through the intersection of this line with the transfer axis AB, draw a line from the assumed value of the stress on the (S) scale to the (d) scale and read the wire diameter (d). On the left portion of the nomograph of Figure 39, draw a line through the values of wire diameter and mean spring diameter on the (d) and (D) scales, respectively, and read the curvature stress correction factor (K) on the (K) scale. Determine the corrected stress by multiplying K by the assumed stress used in the above derivation of wire diameter (d). If the corrected stress is greater than the recommended maximum working stress, the pring must be recalculated using a lower assumed stress in the determination of wire diameter. Also, as stated in paragraph 21.14(b), it is desirable to leave an additional margin whenever practicable so that if the spring were compressed to solid length the corrected stress still would not exceed the allowable working stress. On the nomograph of Figure 40 or 41, connect with a straight line the value of final assembled load (P_2) on the (P) scale with the value of mean spring diameter on the (D) scale. Through the intersection of this line with the transfer axis AB, draw a line from the wire diameter (d) to the (f) scale and read the deflection per coil (f). Values of total number of coils (TC), active coils (N), free length (FL), solid height (SH), and corrected stress at solid height (if pertinent) may be determined in a manner similar to that described in paragraph 21.17.

21.17 EXAMPLE OF COMPRESSION SPRING CALCULATION. A compression spring is required to have the following characteristics:

Work in a $^{13}/_{16}$ inch diameter bore
Final Assembled Length, $L_2 = 1\frac{1}{8}$ in
Initial Assembled Length, $L_1 = 1\frac{5}{8}$ in
Load at $L_2 = P_2 = 25$ lb
Desired Load at $L_1 = P_1 = 20$ lb (tentative)
Frequency of Deflections = 2000 cycles per hr max.
Total Deflections = 500,000 cycles
Ends closed and ground

These additional drawing requirements must be determined:

Material specification
Free length
Diameter of wire
Total coils, REF only

Proceed with the calculation as follows:

Select music wire for the material.

Utilizing Table VI in this calculation, an OD slightly smaller than $^{13}/_{16}$ and a load greater than 25 lb should be selected.

Thus from Table VI, select

OD = .750 in.

Load = 30 lb.

Deflection at 30 lb = .1574 in. per coil

Dia wire = d = .080 in.

Above values are for a stress of 100,000 psi

D = OD — d = .75 — .08 = .67 in.

The clearance between each coil when spring is at final assembled length should be a minimum of 10% of the wire diameter.

Using 10%, d + clearance = 1.1d

Total coils = TC = L_2/1.1d

$$= \frac{1.125}{1.1 \times .08} = 12.8$$

Use TC = 12

Active coils = N = 12 — 2 = 10
Defl/coil/lb = .1574/30 = .00525 in.
Defl/10 coils/lb = .0525 in.

$$\text{Change in Load} = \frac{L_1 - L_2}{.0525}$$

$$= \frac{1.625 - 1.125}{.0525}$$

$$= 9.5 \text{ lb}$$

$$P_1 = P_2 - 9.5 = 25 - 9.5 = 15.5 \text{ lb}$$

This is smaller than the desired load of 20 lb at initial assembled length. If the 15.5 lb ± 1.5 lb load is acceptable, proceed with the computation. If not acceptable, a more flexible spring must be designed. This will require one or more of the following changes:

(a) An increase in the final compressed length, and the same increase in the initial compressed length to allow for additional coils.

(b) An increase in D, possibly accompanied by an increase in d in order to maintain a safe stress.

Assuming in this case that a P_1 of 15.5 lb is satisfactory, proceed as follows:

Free length = FL = 1.125 + (25 × .0525)
= 1.125 + 1.31 = 2.435

Solid height = SH = 12 × .080 = .96

Total Deflection = 2.437 — .96 = 1.477

$$\text{Load at Solid Height} = \frac{1.477}{.0525} = 28 \text{ lb}$$

Stress at 28 lb = 100,000 × 28/30

= 93,500 psi

Stress at 25 lb = 100,000 × 25/30

= 83,500 psi

$$\text{Spring Index} = \frac{D}{d} = \frac{.61}{.08} = 8.37$$

From Figure 39, Correction Factor K = 1.16

Corrected Stress at 28 lb (Solid Height) = 93,500 × 1.16 = 108,000 psi

Since this spring is not designed to deflect to solid height in normal operation, the frequency of such a deflection should not approach the magnitude of even light service. The minimum elastic curve provides the limit for the maximum allowable solid stress.

From Figure 62, the recommended maximum solid stress for .080 music wire is 140,000 psi.

Loads and deflections of helical compression and extension springs

OD of spring			W & M wire size														19	18	17	16	15	14	13	12
			Decimal wire size																					
In.	Dec.		.010	.012	.014	.016	.018	.020	.022	.024	.026	.028	.030	.032	.034	.038	.041	.0475	.054	.0625	.072	.080	.0915	.1055
3/32	.0938	f	.01870	.01564	.01276	.01061	.00893	.00764	.00657	.00570	.00496													
		P	.400	.830	1.351	2.07	3.02	4.26	5.83	7.78	10.18													
7/64	.1094	f	.0277	.0222	.01824	.01529	.01302	.01121	.00974	.00853	.00751	.00664	.00589											
		P	.386	.607	1.130	1.723	2.51	3.52	4.79	6.36	8.28	10.60	13.25											
1/8	.125	f	.0371	.0299	.0247	.0208	.01781	.01548	.01353	.01192	.01055	.00943	.00844	.00758	.00683									
		P	.343	.600	.971	1.475	2.14	2.99	4.06	5.37	6.97	8.89	11.16	13.83	16.95									
9/64	.1406	f	.0478	.0387	.0321	.0273	.0234	.0204	.01791	.01590	.01417	.01271	.01144	.01034	.00937	.00777								
		P	.301	.529	.853	1.291	1.865	2.61	3.53	4.65	6.02	7.64	9.58	11.84	14.47	21.0								
5/32	.1563	f	.0600	.0487	.0406	.0348	.0293	.0261	.0230	.0205	.01832	.01649	.01491	.01354	.01234	.01033	.00909							
		P	.268	.470	.756	1.148	1.656	2.31	3.11	4.10	5.30	6.72	8.39	10.33	12.62	18.22	23.5							
11/64	.1719	f	.0735	.0598	.0500	.0428	.0369	.0324	.0287	.0256	.0230	.0208	.01883	.01716	.01569	.01321	.01172	.00914						
		P	.243	.424	.683	1.031	1.488	2.07	2.79	3.67	4.73	5.99	7.47	9.19	11.19	16.00	21.0	33.8						
3/16	.1875	f	.0884	.0720	.0603	.0516	.0448	.0394	.0349	.0313	.0281	.0255	.0232	.0213	.01940	.01650	.01468	.01157	.00926					
		P	.221	.387	.621	.938	1.351	1.876	2.53	3.32	4.27	.540	6.72	8.27	10.06	14.41	18.47	30.07	46.2					
13/64	.2031	f	.1046	.0854	.0717	.0614	.0534	.0470	.0418	.0375	.0338	.0307	.0280	.0257	.0236	.0201	.01796	.01430	.01155					
		P	.205	.355	.570	.850	1.237	1.716	2.31	3.03	3.60	4.92	6.12	7.52	9.33	13.06	16.99	27.1	41.5					
7/32	.2188	f		.1000	.0841	.0721	.0523	.0558	.0494	.0444	.0401	.0365	.0333	.0306	.0282	.0241	.0216	.01733	.01411	.01096				
		P		.328	.526	.793	1.140	1.580	2.13	2.79	3.58	4.52	5.61	6.88	8.33	11.92	15.22	24.6	37.6	61.8				
1/4	.250	f			.1116	.0950	.0830	.0742	.0663	.0597	.0541	.0494	.0453	.0417	.0386	.0332	.0299	.0242	.01996	.01578	.01234			
		P			.457	.687	.987	1.366	1.834	2.40	3.08	3.88	4.82	5.90	7.14	10.17	12.95	20.6	31.1	51.1	82.4			
9/32	.2813	f			.1432	.1234	.1080	.0958	.0857	.0774	.0703	.0643	.0591	.0546	.0506	.0437	.0396	.0322	.0268	.0218	.01767	.01438		
		P			.403	.606	.870	1.203	1.613	2.11	2.70	3.40	4.22	5.16	6.24	8.86	11.26	18.01	27.2	43.6	70.0	99.9		
5/16	.3125	f				.1542	.1351	.1200	.1075	.0973	.0886	.0811	.0746	.0690	.0640	.0556	.0504	.0418	.0347	.0281	.0225	.01885		
		P				.542	.778	1.074	1.440	1.881	2.41	3.08	3.76	4.60	5.54	7.85	9.97	15.80	23.9	36.3	61.0	84.5		
11/32	.3438	f					.1633	.1470	.1321	.1190	.1090	.0990	.0921	.0862	.0792	.0690	.0627	.0518	.0436	.0346	.0300	.0244	.01982	
		P					.703	.970	1.300	1.697	2.17	2.73	3.38	4.13	4.98	7.06	8.94	14.21	21.3	34.1	53.1	78.2	119.2	
3/8	.375	f						.1769	.1590	.1440	.1314	.1206	.1113	.1031	.0960	.0830	.0764	.0634	.0536	.0438	.0356	.0305	.0247	.01931
		P						.945	1.185	1.444	1.978	2.48	3.07	3.73	4.53	6.40	8.10	12.86	19.27	30.7	48.4	66.2	106.1	171.1

NOTE: For steel wire, f is the deflection of each active coil caused by a load P. Both terms are based on a torsional stress of 100,000 psi and a torsional modulus of 11,500,000 psi.

TABLE VI

Loads and deflections of helical compression and extension springs

OD of spring			W & M wire size																					
									19	18	17	16	15	14	13	12	11	1/8	10	9	8	7	6	5
			Decimal wire size																					
In.	Dec.		.022	.024	.028	.030	.034	.038	.041	.0475	.054	.0625	.072	.080	.0915	.1055	.1205	.125	.135	.1483	.162	.177	.192	.207
25/64	.3906	f	.1733	.1571	.1434	.1216	.1049	.0919	.0836	.0695	.0589	.0483	.0395	.0338	.0275	.0216								
		D	1.135	1.480	1.893	2.94	4.33	6.11	7.74	12.27	18.37	29.2	46.0	64.8	100.5	161.7								
13/32	.4063	f	.1883	.1709	.1560	.1324	.1143	.1001	.0913	.0760	.0645	.0531	.0436	.0373	.0304	.0241								
		D	1.088	1.420	1.815	2.82	4.15	5.85	7.41	11.73	17.56	27.9	43.9	61.6	95.6	153.3								
27/64	.4219	f	.2039	.1851	.1691	.1436	.1242	.1089	.0993	.0828	.0703	.0580	.0477	.0410	.0335	.0266	.0212							
		D	1.046	1.364	1.744	2.71	3.98	5.61	7.10	11.24	16.81	26.7	41.9	58.8	91.0	145.7	228							
7/16	.4375	f	.220	.1999	.1827	.1553	.1343	.1179	.1075	.0898	.0764	.0631	.0521	.0448	.0367	.0293	.0237	.0219						
		D	1.006	1.312	1.678	2.60	3.82	5.39	6.52	10.79	16.13	25.6	40.1	53.3	86.9	138.9	217.	245						
29/64	.4531	f	.237	.215	.1968	.1674	.1449	.1272	.1162	.0971	.0827	.0685	.0566	.0488	.0401	.0321	.0258	.0242						
		D	.970	1.265	1.616	2.51	3.68	5.19	6.57	10.39	15.50	24.5	38.5	53.9	83.2	132.6	207	234						
15/32	.4688	f		.231	.212	.1800	.1560	.1370	.1252	.1048	.0894	.0711	.0614	.0530	.0437	.0351	.0292	.0265	.0232					
		D		1.220	1.559	2.42	3.55	5.00	6.33	9.99	14.91	23.8	37.0	51.7	79.7	126.9	197.3	223	299.					
31/64	.4844	f		.249	.227	.1931	.1674	.1471	.1345	.1127	.0962	.0799	.0663	.0573	.0473	.0382	.0308	.0290	.0259					
		D		1.179	1.506	2.33	3.43	4.83	6.10	9.64	14.37	22.7	35.6	49.7	76.6	121.7	188.8	213	277.					
1/2	.500	f			.243	.207	.1792	.1575	.1441	.1209	.1033	.0859	.0714	.0619	.0512	.0414	.0335	.0316	.0277					
		D			1.456	2.26	3.31	4.67	5.90	9.30	13.87	21.9	34.3	47.0	73.6	116.9	181.1	205	265					
17/32	.5313	f			.276	.235	.204	.1796	.1645	.1382	.1183	.0987	.0822	.0714	.0593	.0432	.0393	.0371	.0327	.0277				
		D			1.366	2.12	3.10	4.37	5.52	8.70	12.06	20.5	31.9	44.6	68.4	108.3	167.3	158.8	244.	335				
9/16	.5625	f				.265	.230	.203	.1961	.1566	.1343	.1122	.0937	.0816	.0680	.0555	.0455	.0430	.0380	.0325	.0278			
		D				1.991	2.92	4.11	5.19	8.18	12.16	19.17	29.9	41.7	63.9	100.9	155.5	175.3	226.	309.	417.			
5/8	.625	f				.331	.288	.254	.233	.1969	.1693	.1420	.1191	.1041	.0873	.0719	.0593	.0561	.0499	.0430	.0371	.0318		
		D				1.782	2.61	3.67	4.63	7.29	10.83	17.04	26.5	36.9	55.4	88.7	136.2	153.4	197.1	269.	360	496.		
11/16	.6876	f					.352	.311	.286	.242	.208	.1753	.1476	.1294	.1089	.0901	.0748	.0710	.0634	.0550	.0475	.0413	.0359	
		D					2.36	3.32	4.19	6.58	9.76	15.34	23.8	33.1	50.5	79.2	121.2	136.3	174.0	[illegible]	317.	420.	561	
3/4	.750	f						.374	.344	.291	.252	.212	.1791	.1574	.1329	.1105	.0923	.0877	.0766	.0685	.0598	.0520	.0455	[illegible]
		D						3.03	3.82	5.99	8.89	13.94	21.6	30.0	45.7	71.5	109.2	122.7	157.1	213.	283.	380.	498.	641.
13/16	.8125	f							.407	.346	.299	.253	.214	.1831	.1594	.1329	.1115	.1081	.0954	.0834	.0733	.0640	.0563	.0497
		D							3.51	5.50	8.15	12.78	19.80	27.5	41.7	65.3	99.3	111.5	142.6	192.9	256	343	448	575

TABLE VI.—(*Continued*)

Loads and deflections of helical compression and extension springs

OD of spring In.	Dec		W & M wire size 17	16	15	14	13	12	11	1/8	10	9
			Decimal wire size .054	.0625	.072	.080	.0915	.1055	.1205	.125	.135	.1483
7/8	875	f	.250	.295	.251	.223	.1882	.1574	.1325	.1262	.1138	.0999
		P	7.83	11.80	18.36	25.3	31.4	50.9	61.1	103.3	130.5	170.3
29/32	9063	f	.277	.320	.271	.239	.204	.1705	.1438	.1370	.1230	.1087
		P	7.28	11.36	17.37	24.3	34.9	57.6	67.5	88.3	125.3	160.0
15/16	9375	f	.405	.344	.292	.256	.219	.1841	.1554	.1479	.1330	.1178
		P	7.00	10.96	16.94	23.5	35.6	55.4	64.1	94.4	120.4	162.3
31/32	9688	f	.435	.369	.313	.277	.236	.1952	.1675	.1508	.1445	.1273
		P	6.76	10.54	16.35	22.6	34.3	53.4	61.0	90.9	115.0	156.1
1	1.000	f	.465	.395	.336	.297	.253	.213	.1801	.1719	.1555	.1372
		P	6.54	10.23	15.80	21.9	33.1	51.8	78.1	87.6	111.7	130.4
1 1/32	1.031	f	.496	.421	.358	.317	.271	.228	.1931	.1843	.1660	.1474
		P	6.33	9.90	15.28	21.1	32.0	49.8	75.8	84.6	107.8	145.1
1 1/16	1.063	f	.520	.448	.382	.338	.289	.244	.207	.1972	.1758	.1580
		P	6.13	9.58	14.80	20.6	31.0	48.3	73.0	81.8	104.3	140.1
1 3/32	1.094	f		.477	.407	.360	.308	.260	.221	.211	.1910	.1681
		P		9.30	14.34	19.83	30.0	46.7	70.0	79.3	100.8	136.5
1 1/8	1.125	f		.507	.433	.383	.328	.277	.235	.226	.204	.1804
		P		9.02	13.92	19.34	29.1	45.2	68.4	76.7	97.0	130.3
1 3/16	1.188	f		.568	.485	.431	.368	.311	.265	.254	.231	.204
		P		8.52	13.14	18.18	27.5	42.8	64.4	72.1	91.7	123.3
1 1/4	1.250	f		.633	.541	.480	.412	.348	.297	.284	.258	.230
		P		8.07	13.44	17.19	26.0	40.3	60.8	68.2	86.6	116.3
1 5/16	1.313	f			.600	.533	.457	.387	.331	.317	.288	.256
		P			11.81	16.31	24.6	38.2	57.7	64.6	82.0	110.1
1 3/8	1.375	f			.662	.588	.506	.429	.367	.351	.320	.285
		P			11.25	15.53	23.4	36.3	54.8	61.4	77.8	104.6
1 7/16	1.438	f			.727	.647	.556	.472	.404	.387	.353	.316
		P			10.73	14.81	22.3	34.0	52.2	58.4	74.1	99.4

OD of spring In.	Dec		5/32	8	7	3/16	6	5	7/32	1/4	9/32	5/16	11/32	3/8
			.1563	.162	.177	.1875	.192	.207	.2188	.250	.2813	.3125	.3438	.375
7/8	875	f	.0928	.0880	.0772	.0787	.0632	.0606	.0542					
		P	200.	234.	312.	377.	407.	521.	626.					
29/32	9063	f	.1010	.0939	.0943	.0773	.0740	.0683	.0606					
		P	190.5	224.	299.	360.	390.	473.	566.					
15/16	9375	f	.1096	.1041	.0917	.0842	.0813	.0723	.0683	.0630				
		P	181.0	215.	296.	343.	373.	477.	572.	803.				
31/32	9688	f	.1183	.1127	.0994	.0913	.0852	.0788	.0721	.0580				
		P	164.5	207.	278.	332.	358.	457.	546.	854.				
1	1.000	f	.1278	.1216	.1074	.0996	.0954	.0853	.0783	.0631	.0615			
		P	177.0	190.8	264.	319.	344.	439.	536.	818.	1216.			
1 1/32	1.031	f	.1374	.1308	.1157	.1065	.1029	.0921	.0845	.0685	.0561			
		P	171.3	191.0	255.	307.	331.	423.	506.	784.	1165.			
1 1/16	1.063	f	.1478	.1404	.1243	.1145	.1107	.0993	.0913	.0740	.0608			
		P	165.6	185.0	246.	296.	319.	407.	487.	755.	1119.			
1 3/32	1.094	f	.1578	.1508	.1333	.1229	.1188	.1066	.0983	.0799	.0637	.0548		
		P	150.9	173.8	238.	286.	300.	393.	470.	727.	1076.	1533.		
1 1/8	1.125	f	.1685	.1604	.1424	.1315	.1273	.1143	.1053	.0850	.0710	.0592		
		P	154.7	173.0	230.	276.	290.	370.	454.	702.	1036.	1474.		
1 3/16	1.188	f	.1900	.1812	.1630	.1486	.1448	.1303	.1203	.0966	.0819	.0687	.0581	
		P	145.4	162.4	215.	250.	270.	336.	424.	655.	965.	1340.	1891.	
1 1/4	1.250	f	.213	.203	.1824	.1690	.1635	.1474	.1363	.1123	.0936	.0780	.0670	
		P	137.0	153.1	203.	244.	263.	334.	390.	616.	902.	1278.	1761.	
1 5/16	1.313	f	.240	.229	.205	.1894	.1834	.1657	.1535	.1268	.1053	.0896	.0766	.0457
		P	129.7	144.7	191.6	230.	248.	315.	378.	577.	848.	1198.	1547.	2208.
1 3/8	1.375	f	.267	.255	.227	.211	.204	.1848	.1715	.1420	.1194	.1014	.0867	.0718
		P	123.0	137.3	181.7	218.	233.	290.	356.	540.	799.	1127.	1518.	2071.
1 7/16	1.438	f	.295	.282	.252	.234	.227	.205	.1906	.1583	.1332	.1136	.0977	.0815
		P	117.0	130.0	172.6	207.	223.	283.	337.	517.	756.	1045.	1459.	1918.

TABLE VI.—(Continued)

Loads and deflections of helical compression and extension springs

OD of spring In.	OD of spring Dec.		3/32	12	11	1/8	10	9	5/32	8	7	3/16	6	5	7/32	4	1/4	9/32	5/16	11/32	3/8	13/32	7/16	1/2
		Decimal wire size	.0938	.1055	.1205	.125	.135	.1483	.1563	.162	.177	.1875	.192	.207	.2188	.2253	.250	.2813	.3125	.3438	.375	.4063	.4375	.500
1 15/32	1.469	f	.865	.494	.423	.405	.370	.330	.310	.306	.305	.280	.258	.318	.300	.1994	.1867	.1607	.1190	.1033	.0896	.0790		
		P	23.6	33.6	51.0	57.1	73.4	97.1	114.2	127.5	166.5	202.	218.	278.	330.	361.	504.	736.	1036.	1410.	1863.	2473.		
1 1/2	1.500	f	.891	.817	.443	.424	.387	.340	.324	.310	.277	.295	.260	.327	.310	.282	.1724	.1682	.1267	.1050	.0947	.0826	.0774	
		P	23.1	33.1	49.8	58.5	70.8	94.8	111.5	124.5	164.6	197.1	213.	260.	321.	353.	461.	717.	1008.	1381.	1841.	2406.	3083.	
1 9/16	1.562	f	.645	.565	.494	.404	.424	.378	.356	.340	.305	.289	.275	.349	.232	.223	.1914	.1535	.1065	.1313	.1065	.0972	.0812	
		P	23.1	31.7	47.7	53.3	67.7	90.6	106.6	118.9	157.1	186.3	263.	257.	306.	336.	464.	682.	958.	1309.	1743.	2279.	2923.	
1 5/8	1.625	f	.791	.614	.537	.505	.461	.413	.387	.370	.333	.309	.300	.273	.264	.244	.218	.1804	.1547	.1330	.1160	.1036	.0905	
		P	24.2	30.3	45.7	51.1	64.8	86.7	102.0	113.0	150.3	180.0	165.0	246.	293.	331.	440.	650.	912.	1246.	1657.	2161.	2788.	
1 11/16	1.688	f	.700	.660	.572	.548	.501	.448	.421	.403	.363	.337	.327	.307	.277	.266	.223	.1971	.1697	.1474	.1290	.1133	.1002	
		P	20.3	29.1	43.6	49.1	63.2	85.2	97.3	108.3	144.1	172.6	185.8	225.	260.	307.	427.	622.	871.	1188.	1577.	2456.	2631.	
1 3/4	1.750	f	.830	.730	.619	.592	.542	.485	.456	.437	.392	.366	.355	.323	.301	.286	.253	.216	.1866	.1613	.1414	.1247	.1105	.0977
		P	19.80	28.0	42.2	47.2	60.6	80.6	94.0	104.9	138.5	165.6	178.4	228.	260.	295.	409.	565.	833.	1125.	1504.	1950.	2504.	3927.
1 13/16	1.812	f	.894	.775	.667	.639	.585	.526	.492	.472	.424	.396	.384	.350	.324	.314	.274	.234	.203	.1761	.1547	.1363	.1212	.0067
		P	18.86	27.0	40.6	45.4	57.6	77.6	90.5	100.8	133.1	150.3	171.4	217.	255.	283.	363.	871.	799.	1066.	1440.	1876.	2391.	3739.
1 7/8	1.875	f	.940	.830	.71.	.667	.620	.564	.530	.506	.457	.436	.414	.377	.351	.330	.296	.253	.219	.1912	.1683	.1493	.1326	.1061
		P	18.20	26.1	39.2	43.0	55.5	74.2	87.2	97.3	138.2	153.4	165.1	209.	248.	272.	370.	548.	767.	1042.	1381.	1793.	2287.	3570.
1 15/16	1.938	f		.802	.768	.736	.676	.605	.560	.546	.498	.458	.446	.405	.379	.365	.330	.273	.237	.207	.1827	.1618	.1442	.1160
		P		25.3	37.8	42.3	53.6	71.6	84.2	93.8	123.6	147.9	158.2	201.	230.	262.	344.	626.	737.	1001.	1323.	1730.	2192.	3414.
2	2.000	f		.956	.823	.780	.723	.649	.610	.585	.537	.492	.478	.435	.407	.393	.314	.295	.256	.224	.1975	.1734	.1566	.1267
		P		34.3	36.6	40.9	51.6	69.2	81.3	93.6	119.4	143.8	153.7	194.3	231.	263.	351.	600.	710.	964.	1274.	1663.	2104.	3273.
2 1/8	2.125	f			.936	.903	.833	.730	.696	.647	.608	.562	.540	.499	.466	.449	.395	.330	.296	.260	.229	.204	.1636	.1482
		P			34.3	38.2	48.5	64.8	76.1	84.9	111.8	133.6	142.8	181.6	210.	236.	327.	476.	661.	890.	1183.	1532.	1948.	3021.
2 1/4	2.250	f			1.056	1.013	.930	.825	.787	.746	.681	.637	.610	.546	.520	.511	.449	.357	.337	.296	.263	.255	.311	.1718
		P			32.3	36.1	45.7	60.9	71.8	79.8	106.7	123.5	135.0	170.5	202.	222.	307.	446.	618	837.	1605.	1430.	1814.	2705.
2 3/8	2.375	f			1.104	1.136	1.043	.930	.904	.840	.763	.716	.666	.637	.606	.576	.507	.437	.383	.337	.300	.266	.341	.1972
		P			30.5	34.1	43.1	57.6	67.8	75.3	99.1	115.3	127.3	160.7	190.7	209.	280.	417.	581.	786.	1036.	1336.	1697.	2613.
2 1/2	2.500	f				1.206	1.103	1.044	.906	.940	.865	.800	.778	.713	.667	.644	.506	.491	.430	.370	.338	.303	.373	.224
		P				32.3	40.9	54.5	64.0	71.3	93.7	111.8	130.4	151.6	180.2	197.5	273.	294.	543.	741.	976	1259.	1594.	2455

TABLE VI.—(*Continued*)

Loads and deflections of helical compression and extension springs

OD of spring			W & M wire size																					
			9	5/32	8	7	3/16	6	5	7/32	4	3	1/4	2	9/32	0	5/16	2-0	11/32	3/8	13/32	7/16	1/2	9/16
			Decimal wire size																					
In.	Dec.		.1483	.1563	.162	.177	.1875	.192	.207	.2188	.2253	.2437	.250	.2625	.2813	.3065	.3125	.331	.3438	.375	.4063	.4375	.500	.5625
2 3/4	2.750	f	1.200	1.208	1.130	1.049	.983	.956	.876	.831	.794	.725	.701	.682	.608	.547	.534	.496	.478	.432	.370	.348	.284	.239
		P	40.3	57.8	64.4	84.6	101.0	108.6	137.0	163.4	177.9	227.	246.	286.	354.	463.	491.	509.	603.	872.	1124.	1432.	2182.	3193.
2 7/8	2.875	f	1.408	1.327	1.274	1.154	1.051	1.052	.985	.904	.874	.797	.773	.730	.671	.604	.590	.549	.523	.468	.421	.381	.317	.287
		P	47.0	55.1	61.7	80.7	96.3	103.6	130.5	154.8	169.5	210.	234.	272.	337.	440.	467.	500.	631.	828.	1067.	1369.	2067	3022.
3	3.000	f		1.433	1.394	1.263	1.194	1.152	1.067	.992	.980	.874	.849	.801	.737	.664	.649	.604	.576	.515	.466	.421	.351	.298
		P		53.7	58.7	77.1	93.0	99.0	121.7	147.6	161.8	200.	223.	260.	322.	420.	446.	534.	601.	789.	1016.	1253.	1961.	2867
3 1/8	3.125	f			1.530	1.378	1.291	1.257	1.154	1.083	1.047	.955	.927	.878	.807	.737	.710	.662	.631	.566	.510	.463	.387	[illegible]
		P			56.3	73.6	89.1	94.8	119.4	141.5	154.9	197.2	216.	248.	307.	401.	426.	510.	574.	753.	969.	1223.	1870	[illegible]
3 1/4	3.250	f				1.407	1.401	1.366	1.255	1.178	1.139	1.040	1.010	.956	.879	.793	.775	.723	.690	.618	.558	.507	.424	[illegible]
		P				70.6	84.5	90.9	114.5	135.6	148.3	180.0	206.	238.	291.	381.	406.	488.	549.	720.	928.	1160.	1793	[illegible]
3 3/8	3.375	f				1.621	1.520	1.499	1.360	1.277	1.235	1.128	1.096	1.036	.955	.862	.842	.785	.750	.673	.609	.553	.461	.395
		P				68.1	81.2	87.3	109.9	130.3	143.6	181.5	196.4	228.	283.	369.	39.1	468.	527.	690.	857.	1119.	1706	2185.
3 1/2	3.500	f					1.643	1.589	1.466	1.379	1.336	1.220	1.185	1.120	1.033	.933	.912	.851	.813	.731	.661	.602	.505	.431
		P					76.1	84.0	106.8	125.3	137.2	174.5	189.0	219.	273.	354.	376.	449.	508.	663.	851.	1073.	1634	2370.
3 5/8	3.625	f					1.768	1.723	1.593	1.487	1.430	1.316	1.278	1.200	1.115	1.008	.986	.919	.878	.790	.715	.651	.548	.464
		P					75.3	81.0	101.9	120.7	133.1	166.1	181.9	211.	261.	341.	36.2	432.	466.	637.	818.	1031.	1571	2282
3 3/4	3.750	f						1.850	1.701	1.590	1.548	1.416	1.374	1.301	1.200	1.085	1.061	.991	.946	.853	.772	.704	.592	.507
		P						76.1	96.3	116.6	127.4	163.1	175.4	204.	253.	328.	319.	417.	460.	614.	788.	993.	1511.	2153.
4	4.000	f							1.950	1.833	1.774	1.694	1.578	1.493	1.379	1.249	1.221	1.141	1.091	.983	.892	.814	.697	.590
		P							91.8	108.7	119.0	151.3	163.7	190.0	235.	306.	325.	358.	437.	571.	733.	923.	1403	2033.
4 1/2	4.500	f									2.28	2.08	2.03	1.920	1.775	1.610	1.574	1.473	1.400	1.273	1.157	1.058	.898	.770
		P									105.1	133.5	144.4	167.6	207.	270.	286.	342.	384.	503.	643.	809.	1227.	1776.
5	5.000	f											2.53	2.40	2.22	2.02	1.973	1.847	1.709	1.600	1.487	1.335	1.136	.942
		P											120.2	140.9	155.3	211.	256.	306.	343.	446.	573.	721.	1001.	1575.
5 1/2	5.500	f													2.72	2.47	2.43	2.26	2.17	1.985	1.793	1.640	1.403	1.210
		P													167.5	218.	231.	270.	310.	404.	517.	649.	902	1416.
6	6.000	f															2.90	2.72	2.61	2.35	2.16	1.904	1.697	1.470
		P															211.	231.	282.	368	471.	591.	893	1743

TABLE VI.—(*Continued*)

Hence, the corrected stress at solid height is less than the recommended minimum elastic limit and is satisfactory.

Corrected stress at 25 lb=83,500 × 1.16 =97,000 psi

From paragraph 28, Average Service would cover 500,000 cycles at a frequency not exceeding 2000 cycles per hour.
From Figure 62, the recommended maximum working stress for .080 music wire under average service is 112,000 psi. Therefore the design is satisfactory for the corrected design stress of 97,000 psi

$$\text{From Table III, } p = \frac{FL - 2d}{N} = \frac{2.437 - 2 \times .08}{10} = .228$$

From paragraph 21.3:

$$OD_c = \sqrt{D^2 + \frac{p^2 - d^2}{\pi^2}} + d$$

$$= \sqrt{.67^2 + \frac{.228^2 - .08^2}{\pi^2}} + .08$$

$$= .753$$

From Table XVI, tolerance on OD= ± .015
Hence, Max OD=.753 + .015=.768, which is satisfactory in an .812 dia bore.

The requirements have been met by a spring having the following characteristics:

OD = .75 d = .08, N = 10, TC = 12, FL = 2.437 (P_1 load at initial assembled length having been changed to 15.5).

21.17.1 Necessity for Several Calculations. Frequently the first set of calculations does not result in a satisfactory design for the conditions involved. It is usually necessary to make several sets of calculations before determining the final design. This often is caused by a stress that is too high, a difficult index for manufacture, or a length that would buckle and require support in a tube or over a rod.

21.18 STRANDED WIRE HELICAL COMPRESSION SPRINGS.

21.18.1 General. Helical compression springs made from standard wire have an inherent tendency about twice as high as round wire springs to dampen high velocity displacement of their coils under shock loading. Under such conditions of loading, they have withstood 3 to 4 times as many deflections as round wire springs having the same load-deflection and stress conditions, before failure. For this reason, they have been used for machine guns. They do not have longer life than round wire springs under normal types of load applications. Good results have been obtained by using three strands of music wire twisted so that the ratio of length of lay to the strand diameter is between 5 and 5½. The length of lay is the distance parallel to the strand axis in which a single wire makes one turn. Preforming the wire by twisting it slightly just prior to the actual stranding operation helps to keep the strands tightly together. Corrosion resisting steel wire also could be used, Springs with an index D/d of 13 can be coiled on automatic spring coilers, but springs with a smaller index usually require coiling over an arbor. Shot peening is not recommended as the small shot lodges tightly between the strands and is difficult to detect and remove.

21.18.2 Stress. High stresses are used in design. The following stresses have been used at solid length; for music wire .030 in. dia 170,000 to 195,000 and for music wire .070 in. dia 150,000 to 170,000 psi.

21.18.3 Formulas for 3 Stranded Wire Spring Design Follow: The S_t and d values are for each strand of wire, and P is the actual load on the spring.

$$P = \frac{G\,d^4\,F}{2.54\,D^3\,N}$$

$$S_t = \frac{G\,d\,F}{\pi\,N\,D^2}$$

$$N = \frac{G\,d^4\,F}{2.54\,P\,D^3}$$

$$r = \frac{P}{F}$$

The design procedure is the same as for other types of compression springs except that each strand carries its proportionate share of the load.

22. EXTENSION SPRINGS

22.1 DEFLECTION OF EXTENSION SPRING ENDS. Loading an extension spring having hook (loop) ends causes the hooks to deflect. The amount of this deflection depends on the type of hook used. For a half hook the deflection per hook is equivalent to .1 of a full coil and the total number of active coils for design purposes will be N + .2. When a full hook is turned up from a full coil, the deflection per hook is equivalent to .5 of a full coil and the total number of active coils for design purposes will be N + 1.

22.2 STRESSES IN HOOKS OF EXTENSION SPRINGS. The hooks at the ends of extension springs are subjected to both tension (bending) and torsional stresses. See Figure 43. These combined stresses are frequently the limiting factor which determines the characteristics of the spring. These stresses occur at the base of the hooks and their magnitude is higher than the stress in the body. Therefore this is the weakest point in an extension spring and the stresses shoud be calculated. The allowable working stresses should not exceed those shown in the curves, Figure 62, Section II of this Appendix, if long life is required.

FIBER STRESS (Not Corrected for Curvature) vs. LOAD
Helical Extension and Compression Springs

Low Range {Mean Diameter: .1" to 3"
Wire Diameter: .015" to .5"}

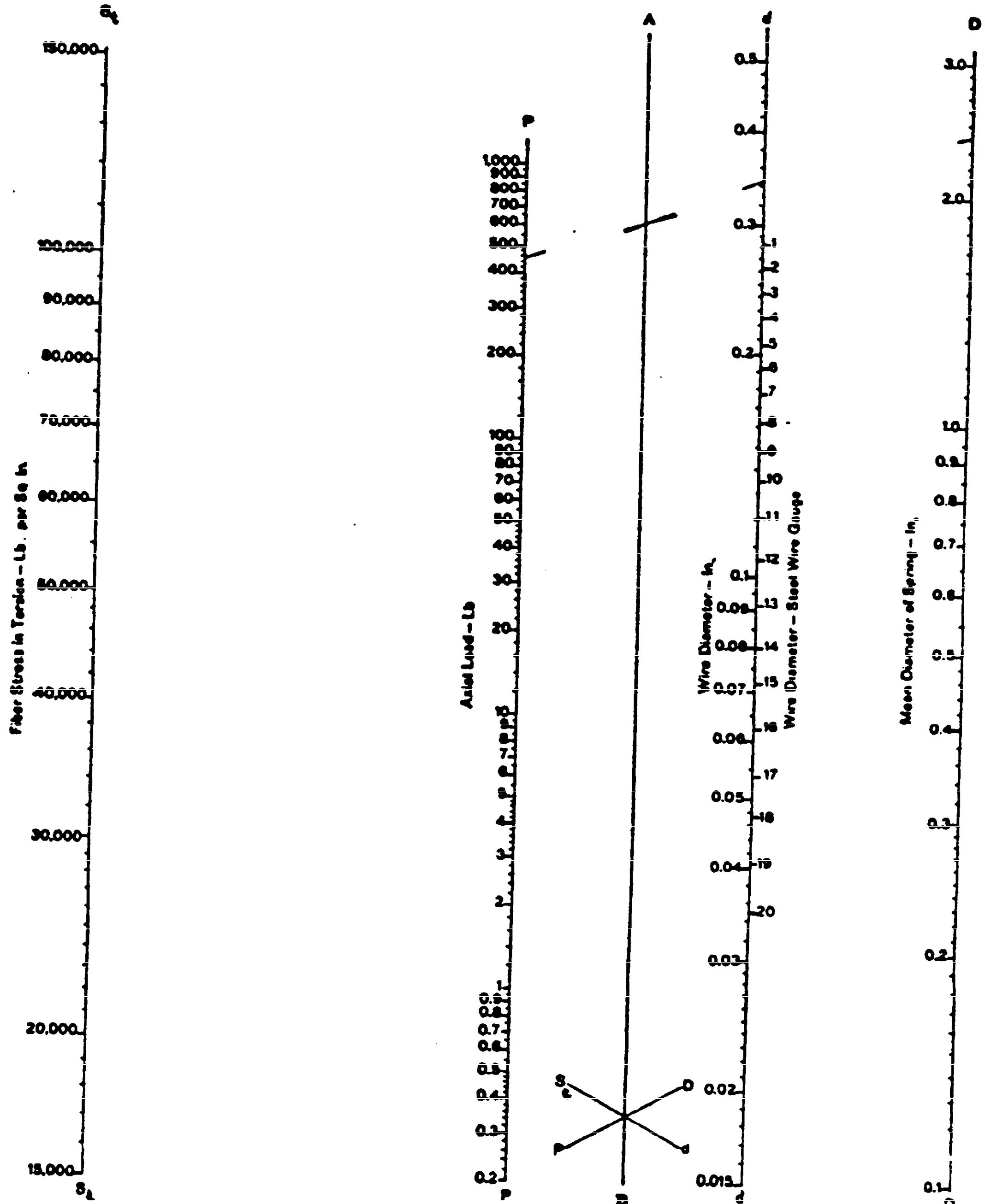


FIGURE 37

FIBER STRESS (Not Corrected for Curvature) vs. LOAD
Helical Extension and Compression Springs

High Range {Mean Diameter: 1" to 10"
Wire Diameter: .15" to 2.5"}

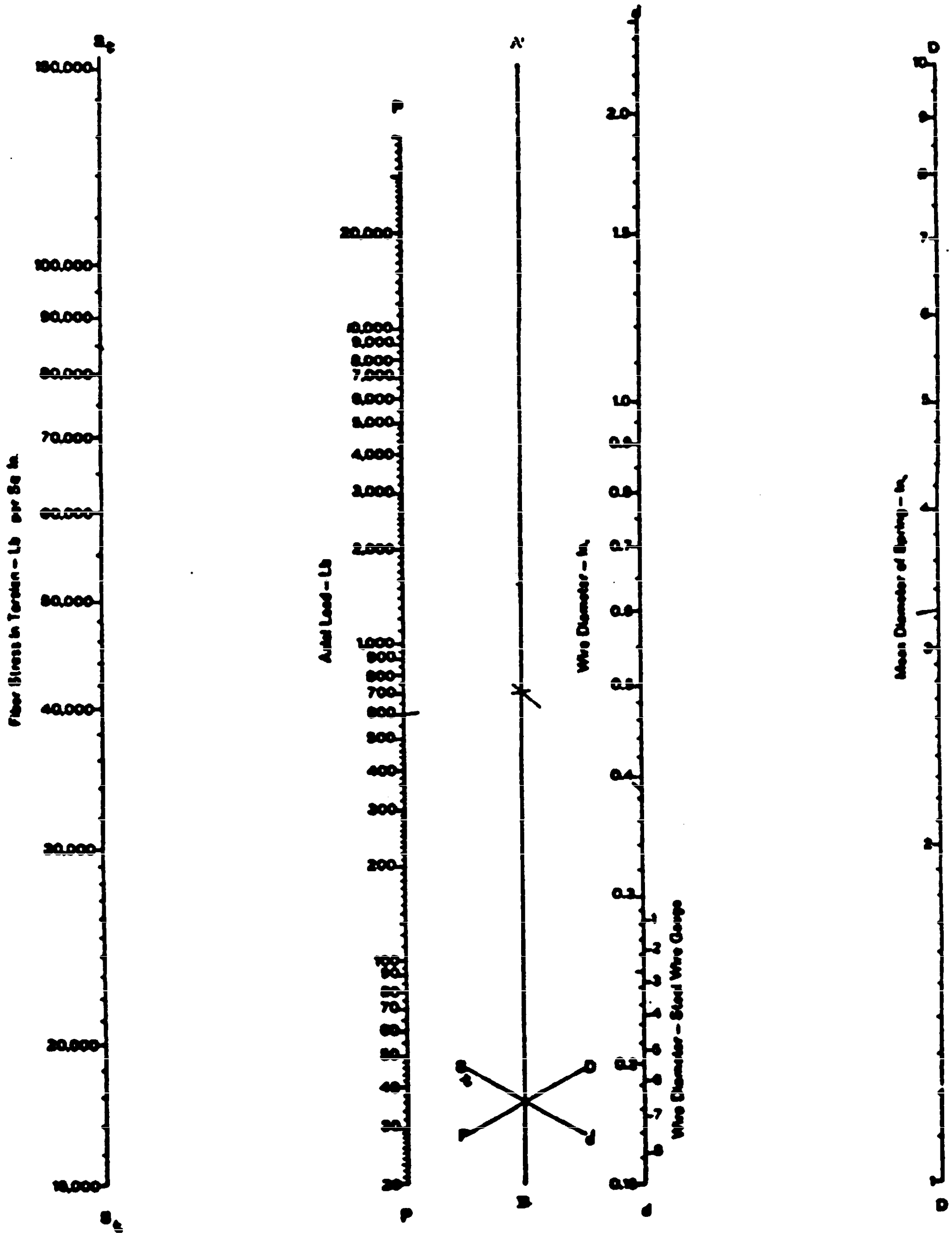


FIGURE 38

FIBER STRESS CORRECTION FOR CURVATURE

Helical Extension and Compression Springs

Find Correction Factor From Spring Index

Using Correction Factor Found on Left Half of Chart — Determine True Fiber Stress

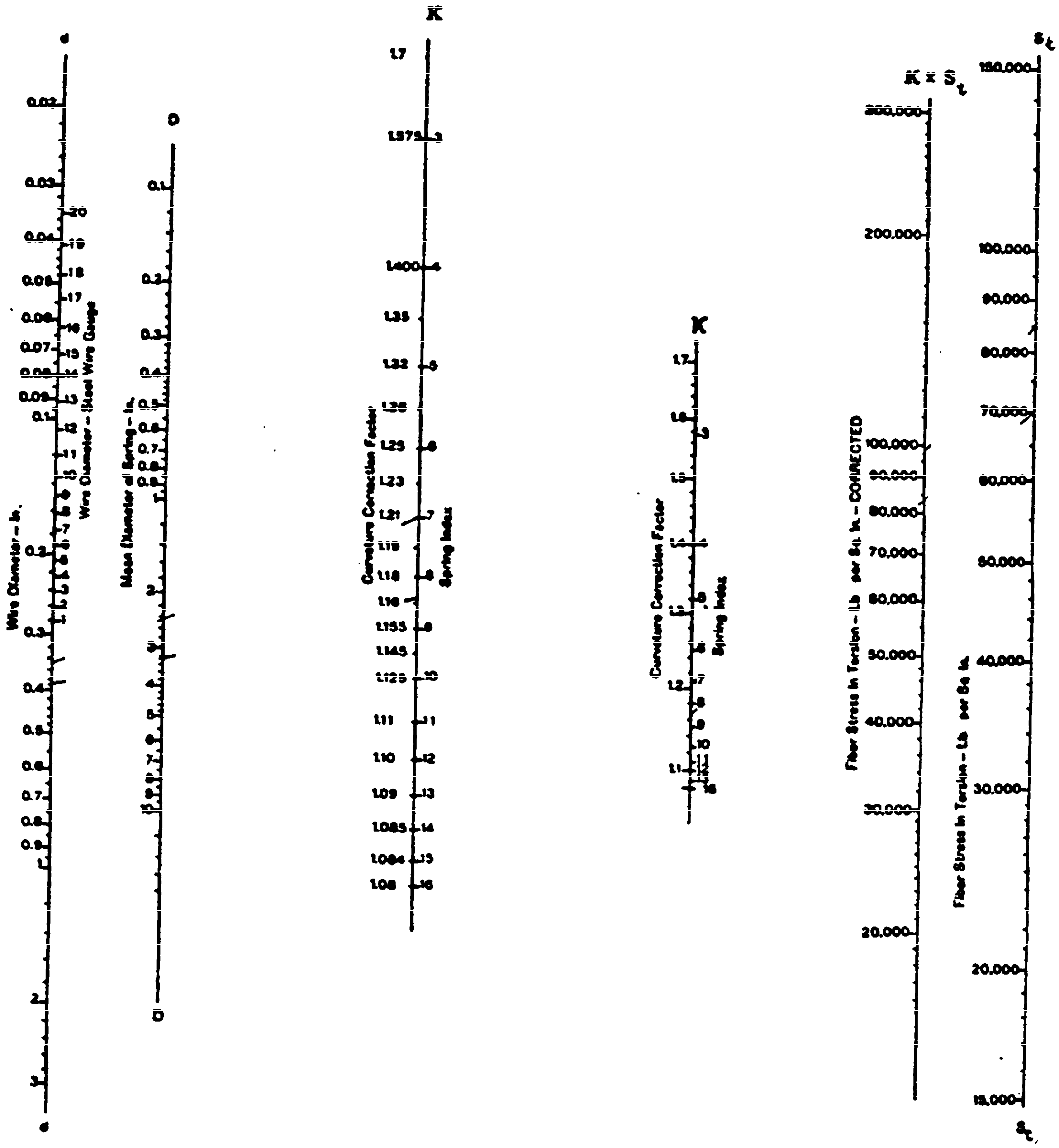


FIGURE 39

DEFLECTION PER COIL vs. LOAD
Helical Extension and Compression Springs

Low Range { Mean Diameter: .1" to 3"; Wire Diameter: .015" to .5" }

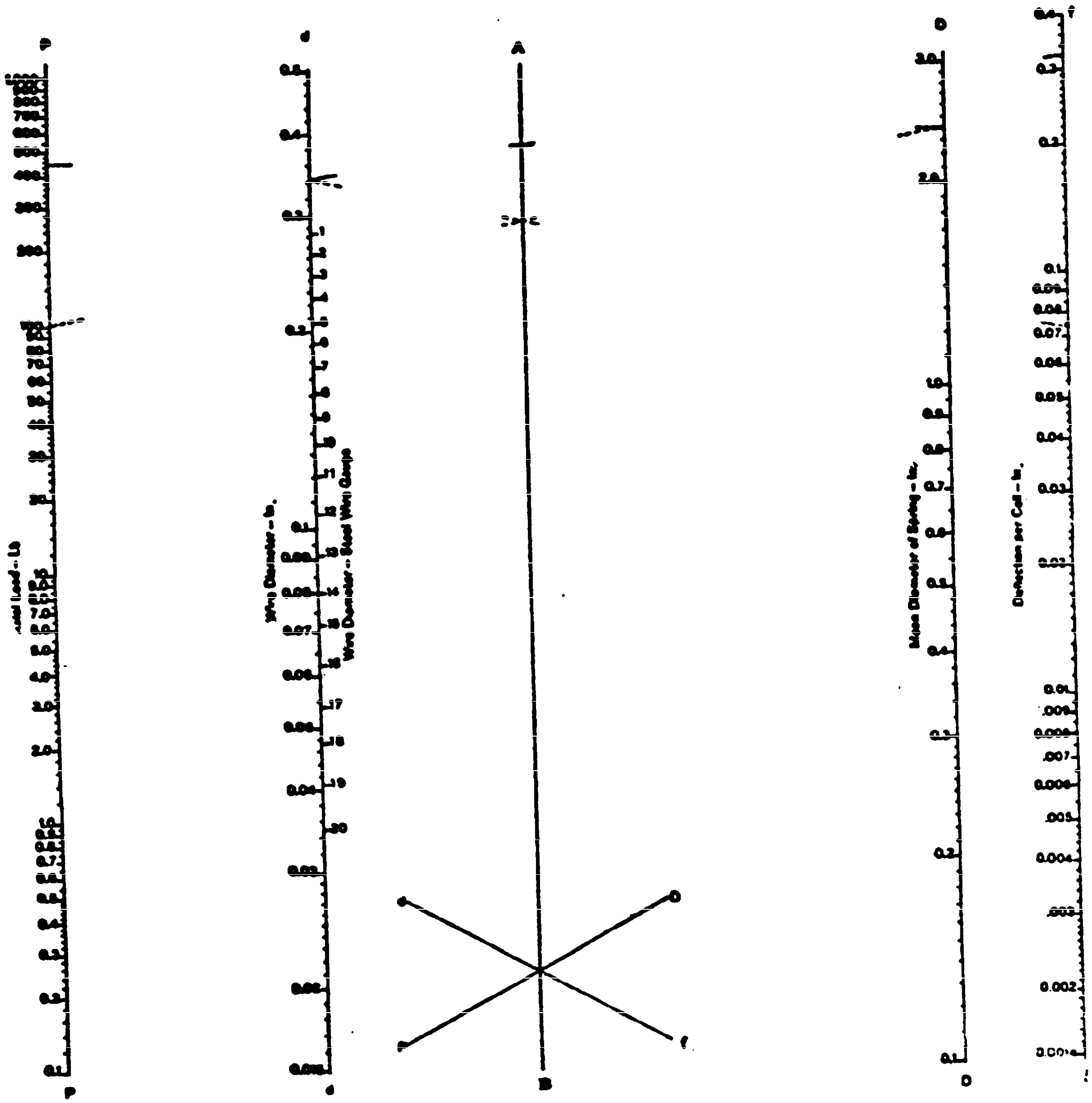


FIGURE 40

DEFLECTION PER COIL vs. LOAD
Helical Extension and Compression Springs

High Range {Mean Diameter: 1" to 10"
Wire Diameter: .15" to 2"

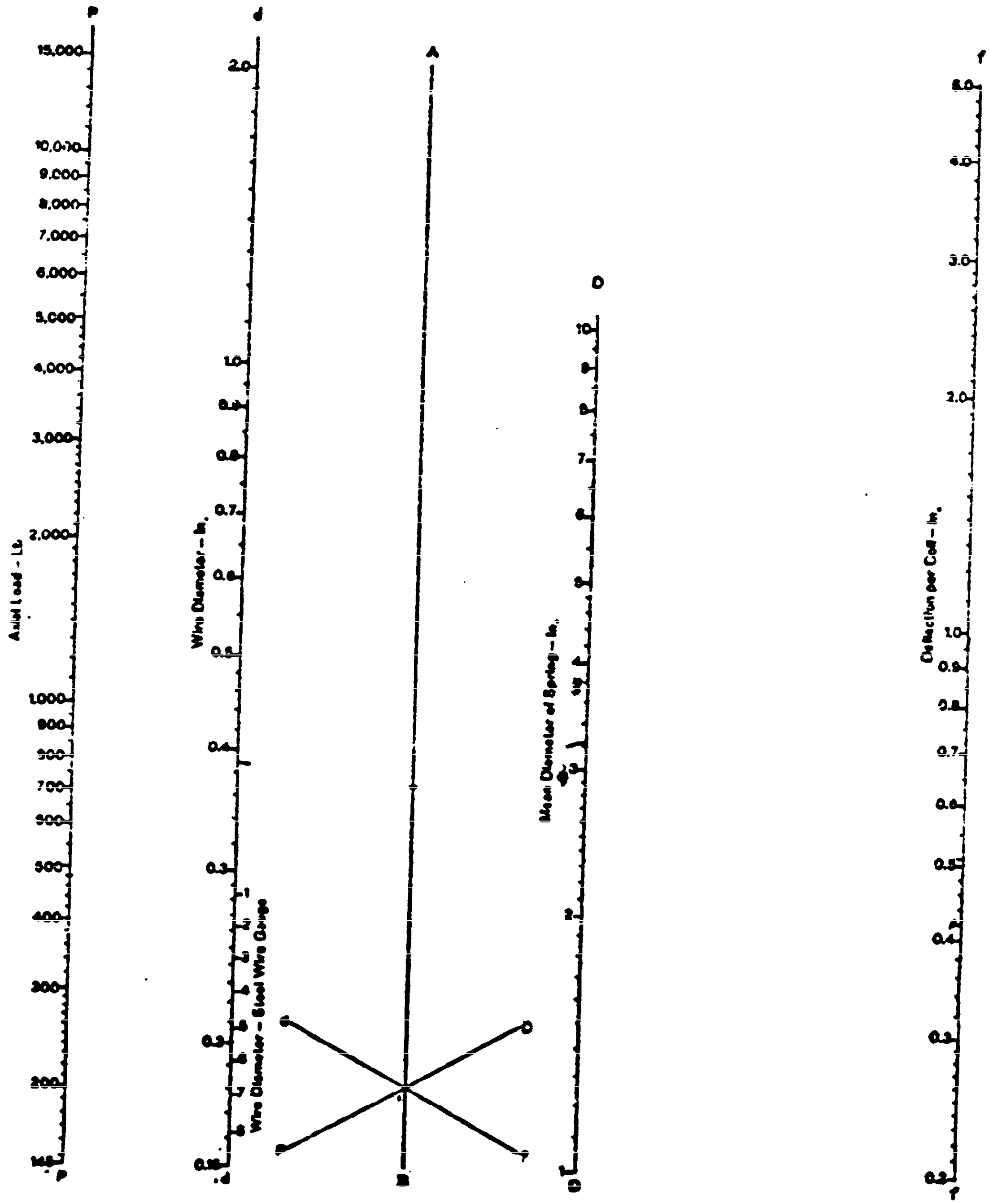


FIGURE 41

DEFLECTION PER COIL

Modulus (G) Other Than 11.5 × 10^6

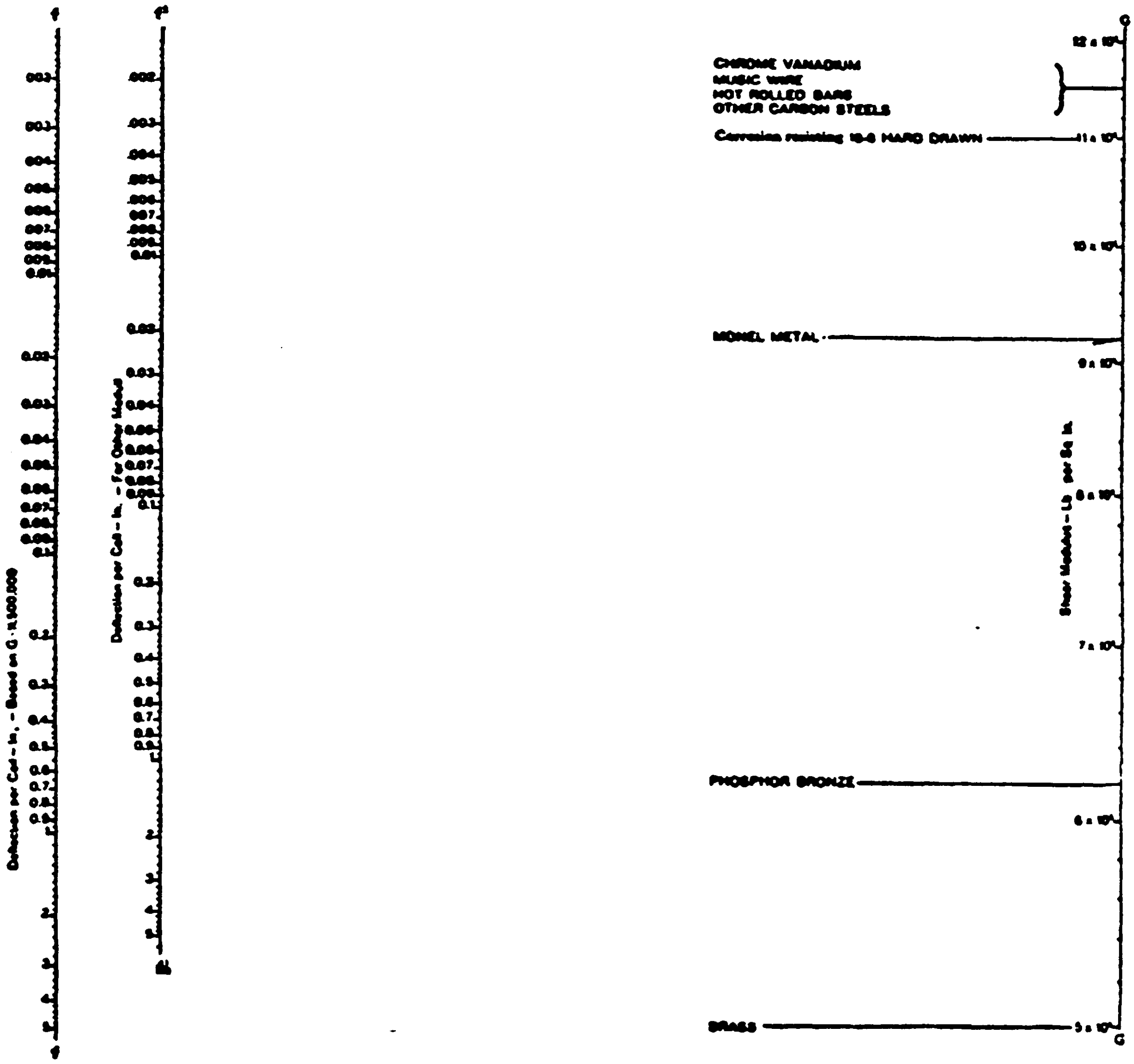


FIGURE 42

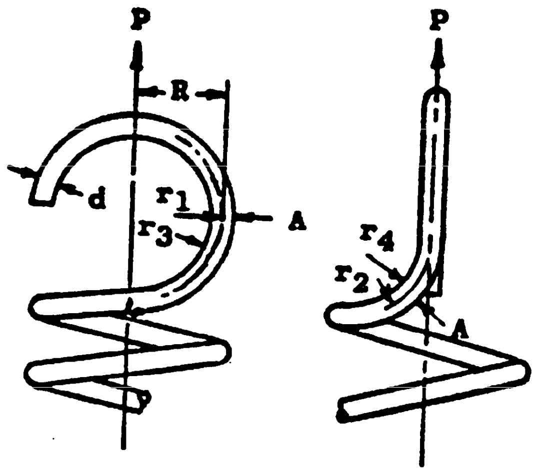


FIGURE 43

Bending Stress at Section A

$$S_b = \frac{PR}{.098\ d^3} \times \frac{r_1}{r_3}$$

Torsional Stress at Section A′

$$S_t = \frac{16\ P\ R}{\pi\ d^3} \times \frac{r_2}{r_4}$$

Whereff:

r_1=Mean Radius of Hook, in.
r_2=Mean Radius of Bend, in.
r_3=Inside Radius of Hook, in.
r_4=Inside Radius of Bend, in.

For best results the inside radius should be at least twice the wire diameter. Special ends can be used when high stresses occur in the hooks. By using a smaller diameter for the last few coils, see Figure 44, before the loop, the magnitude of PR is reduced. Thus the stress is reduced in direct proportion to the decrease in the magnitude of PR. By using as large radii for r_3 and r_4 as the design will permit the stress value is further reduced. The values of r_3 and r_4 can be determined by layout.

In extension springs with hooks bent off the body (see Figure 6, Off-set hook at side) the moment arm of the load on which the maximum torsion stress in the spring depends is about twice what it would be if the load were applied axially. This means doubling the stress for a given load.

22.3 INITIAL TENSION. Initial tension is a load in pounds which opposes the opening of the coils by an external force. It is wound into the springs during the coiling operation. Extension springs will have a uniform rate after the applied load overcomes the load due

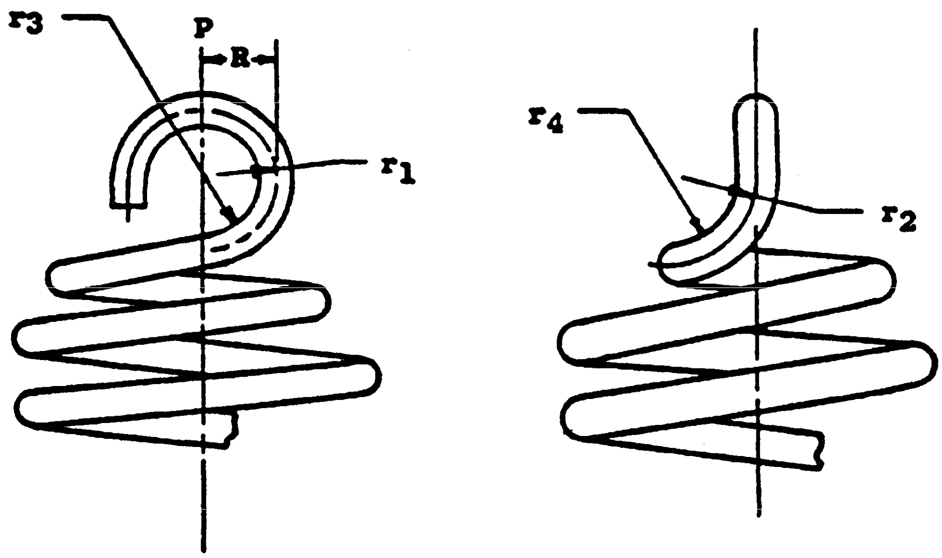


FIGURE 44

to initial tension. The number of coils do not affect the amount of initial tension except when the weight of the coils is heavier than the initial tension. The amount of initial tension is dependent on the spring index (D/d); the smaller the index the larger the initial tension. Initial tension does not increase the ultimate load or capacity of the spring but causes a larger portion thereof to be exerted during the initial deflection. For example, if the initial tension is 4 lbs and the spring rate is 9 lb then, at 1 inch deflection the load is

$(1 \times 9) + 4 = 13$ lb

3-inches deflection the load is

$(3 \times 9) + 4 = 31$ lb

In computing the total torsional stress add the torsional stress caused by initial tension to the torsional stress caused by deflection. Figure 45 shows the amount of initial tension in terms of torsional stress (without application of curvature stress correction factor) which can be coiled into extension springs made of music wire, oil tempered, corrosion resisting steel and hard drawn spring steels. Reduce these values 20 percent for springs made from nickel-base alloys such as Monel and Inconel. Hot rolled springs and those made of annealed materials cannot be wound with initial tension. Springs which require stress relieving will lose 25 to 50 percent of their initial tension. This loss can be compensated for during the coiling operation by winding more initial tension into the spring and thus obtain the required initial tension after stress relieving.

22.4 EXAMPLE OF EXTENSION SPRING CALCULATION. An extension spring is required to exert a force (load) of 27 lb at 2 in. deflection and be deflected an additional 3 in. (5 in. total) and then exert a total load of 55 lb. It must operate within a 1.812 in. minimum diameter bore.

Select a suitable oil tempered wire diameter and determine the number of coils, length over coils, free length inside ends, maximum extended length inside ends without set, stresses at all loads, etc., stressed for light service (see Figure 62)

From Table VI it will be found that a spring with an OD of 1¾ in., and made from 5/32 in. (0.156 in.) diameter wire, will exert a force (load) of 94.0 lb at a stress of 100,000 psi and have a deflection per coil of 0.456 in. at that load.

Stress at 55 lb $= \frac{55}{94} \times 100{,}000$

$= 58{,}500$ psi

Rate or load per inch, $\frac{55\text{-}27}{3} = 9.33$ lb/in,

Load due to deflection of 2 in.,
$2 \times 9.33 = 18.66$ lb

Load required is 27 lb; therefore the initial tension is $27 - 18.66 = 8.34$ lb

Load at 5 in. deflection $= (5 \times 9.33) + 8.34 = 46.65 + 8.34 = 54.99$ (say 55 lb)
Stress at 5 in. deflection (due to 46.65 lb load) =

$$\frac{46.65}{94} \times 100{,}000 = 49{,}650 \text{ psi}$$

Stress due to initial tension,

$$\frac{8.34}{94} \times 100{,}000 = 8{,}880 \text{ psi}$$

Deflection per coil, $\frac{49{,}650}{100{,}000} \times$

$0.456 = 0.226$ in.

Number of active coils for 5 in. deflection.

$$\frac{5}{.226} = 22.1 \text{ (say 22)}$$

Length over coils, $(22 + 1) \times 0.156 = 3.58$ (say 3-9/16 in.)

Length of hook (assuming 80 per cent of ID), $0.80 \times 1.437 = 1.15$ (say, 1 5/32 in.)

Free length, inside ends (hooks), 3 9/16 + (2 × 1 5/32 in.) = 5⅞ in.

The formulas for solving the example follow:

Stress due to 46.65 lb load.

$$S_t = \frac{PD}{0.393d^3} = \frac{46.65 \times 1.594}{0.393 \times 0.156^3}$$

49,650 psi

$$N = \frac{GDF}{3.14 S_t D^2}$$

$$= \frac{11{,}200{,}000 \times 0.156 \times 5}{3.14 \times 49{,}650 \times 1.594^2} = 22.1$$

Stress due to initial tension, =

$$S_{it} = \frac{S_t}{P} \times IT = \frac{49{,}650}{46.65} \times 8.34$$

= 8,870 psi

Final stress, 49,650 ± 8,870

=58,520 psi

$$\text{Spring Index} = \frac{1.594}{.156} = 10.2 \text{ From}$$

Figure 39, Correction Factor

=1.12

Corrected torsion stress in coils at 55 lb load

=53,520 × 1.12=65,200 psi

Since the allowable stress from Figure 62= 92,000 × .85=78,200 psi, design is satisfactory so far.

Torsional stress in hooks at 55 lb load=

$$\frac{16PR}{\pi d^3} \times \frac{r_2}{r_4} = \frac{16(55)\ .797}{\pi(.00380)}$$

$$\times \frac{.797}{.719} = \frac{700}{.01196} \times 1.11 = 65{,}000 \text{ psi}$$

Bending stress in hooks at 55 lb load

$$= \frac{PR}{.098d^3} \times \frac{r_1}{r_3} = \frac{55 \times .797}{.098 \times .156^3}$$

$$\times \frac{.797}{.719} = 130{,}000 \text{ psi}$$

Since the allowabe stress from Figure 62= 92,000 × 1.5 = 138,000 psi, the present hook configuration is satisfactory.

To determine the maxium extended length inside ends without permanent set we determine (Figure 62) the minimum elastic limit for .156 dia oil tempered wire to be 100,000 × .85 = 85,000 psi.

Allowing for curvature stress correction factor of 1.12, the allowable stress =

$$\frac{85{,}000}{1.12} = 75{,}800$$

From Table VI, we know that a load of 94 lb will produce a stress of 100,000 psi. Therefore, the load producing 75,800 psi is as follows:

$$\frac{\times}{94} = \frac{75{,}800}{100{,}000};\ \times = \frac{94(75{,}800)}{100{,}000} = 71 \text{ lb}$$

Since load at 5 in. = 55 lb and rate = 9.33 lb/in., a 71 lb load would occur at

$$\frac{71\text{-}55}{9.33} + 5 \text{ in.} = \frac{16}{9.33} + 5 = 1.715 + 5 =$$

6.715 in. (say 6⅝ in.) This length is tentative, depending on the following stress checks.

Checking the torsion stress in the hooks at maximum extended length (load=71 lb):

$$S_t = \frac{16PR}{\pi d^3} \times \frac{r_2}{r_4} = \frac{16 \times 71 \times .797}{\pi(.00380)}$$

$$\times \frac{.797}{.719} = 84{,}000 \text{ psi}$$

Since the allowable torsional stress from Figure 62 = 100,000 × .85 = 85,000 psi, the hooks torsional stress is satisfactory.

Checking the bending stress in the hooks at maximum extended length (load = 71 lb)

$$S_b = \frac{PR}{.098d^3} \times \frac{r_1}{r_3}$$

$$= \frac{71 \times .797}{.098 \times .156^3} \times \frac{.797}{.719}$$

= 168,000 psi

Since the allowable bending stress at maximum extended length = 102,000 × 1.50 = 153,000 psi; the hook is overstressed and the maximum extended length must be decreased. Therefore, we calculate the maximum load without set.

$$\frac{153{,}000}{168{,}000} = \frac{X}{71};\ X = \frac{71\,(153{,}000)}{168{,}000}$$

$= 64.7$ lb

Since 55 lb 5 in., with rate of 9.33 lb/in., a 64.7 lb load would occur at

$$\frac{64.7\text{-}55}{9.33} + 5 \text{ in.} = \frac{9.7}{9.33} + 5 = 1.04 + 5 =$$

6.04 in.

The maximum extended length inside ends without permanent set is: 6.04 + 5.875 = 11.915, say 11.90.

PERMISSIBLE TORSIONAL STRESS RESULTING FROM INITIAL TENSION IN COILED EXTENSION SPRINGS FOR DIFFERENT D/d RATIOS

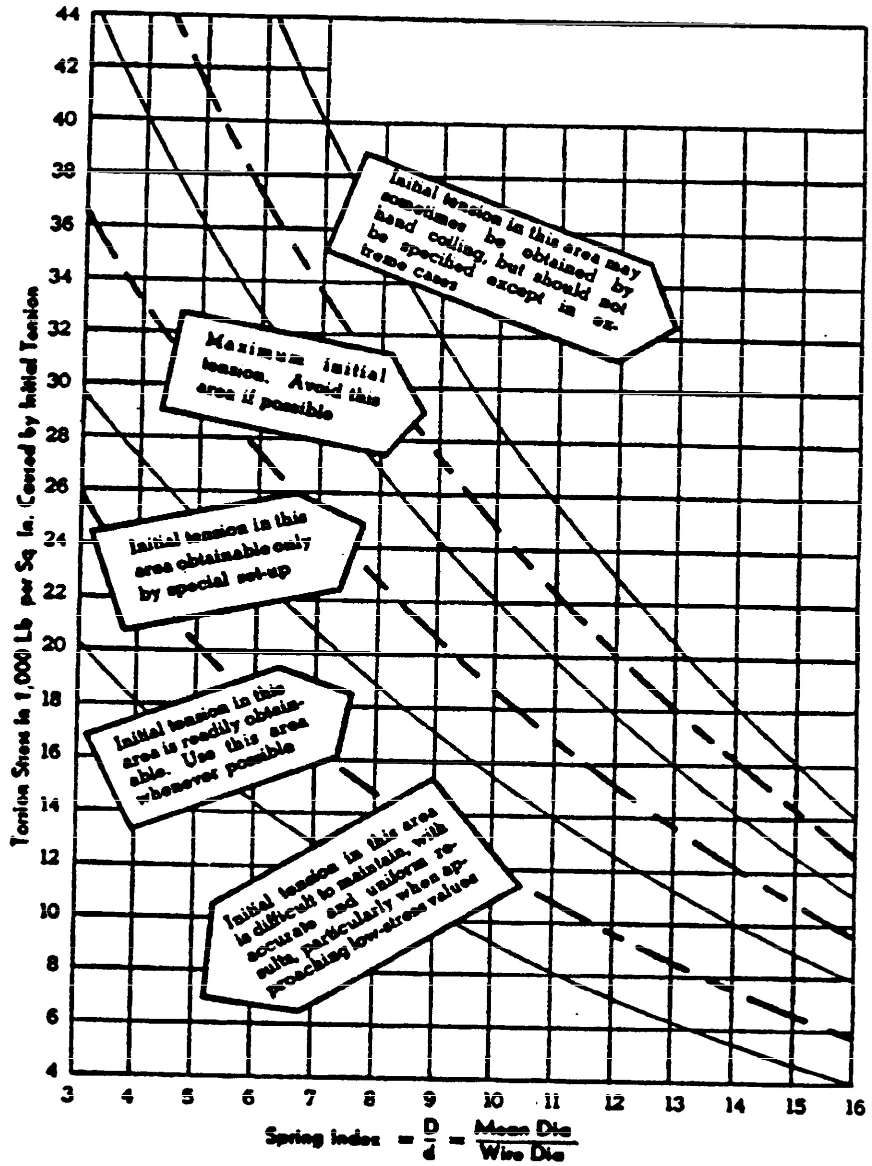


FIGURE 45

22.5 PRECAUTIONS AND SUGGESTIONS FOR EFFECTIVE DESIGN OF EXTENSION SPRINGS.

(a) Avoid using enlarged, extended, or specially shaped hooks or loops; they may double the cost of the spring and have high stress concentrations.

(b) If a plug must screw into the end of a spring, the spring should be coiled right hand.

(c) Nearly all extension springs are wound with enough initial tension to keep the spring together. Always figure on at least 5 to 10 per cent of the final load as initial tension, unless otherwise specified.

(d) Electroplating does not deposit a good coating on the inside of, or between, the coils of extension springs.

(e) Hooks on extension springs deflect under a load. Each half hook, made by bending one-half of a coil, deflects an amount equivalent to 0.1 of an active coil. Each full hook is equivalent to 0.5 of an active coil. Allowance for this deflection should be considered in design.

(f) If the relative position of the ends is not important note this fact on the drawing.

(g) For standard hooks keep the OD of the hook the same as the OD of the spring, and the distance from the end of the body, or from the last coil, to the inside of the hook about 75 to 85 per cent of the ID of the spring.

(h) The body length or closed portion of an extension spring equals the number of coils in the body plus one, multiplied by the wire diameter.

(i) When deflected 1¼ times the maximum deflection as assembled, the total stress should be less than the Minimum Elastic Limit shown by the curves in Figure 62, as modified by their multiplying constants.

22.6 GARTER SPRINGS

22.6.1 General. Close coiled extension springs used in the form of rings by connecting the ends are often used as driving belts, for oil seals and as retainers. The ends may have half or full loops and then be hooked together or one end may be reduced in diameter for three to six coils and screwed into the other end. Connecting the ends with a separate short section, called a connector, is occasionally done.

22.6.2 Formulas. The following formulas for design purposes may be used:

$$\text{Deflection, } F = \pi\,(\text{Shaft Diameter} + \text{OD of spring}) - (FL)$$

$$\text{Rate} = r = \frac{G\,d^4}{8\,N\,D^3}$$

Pressure per inch of circumference on shaft for spring with initial tension, equals

$$2\,\pi r - \frac{2\,\pi\,(\text{ID of Connected Ring})\,r}{\text{Shaft dia}} + \frac{2\,(IT)}{\text{Shaft dia}}$$

(Wherein ID of Connected Ring equals π (FL—OD)

Pressure of each coil on the shaft equals the pressure per inch of circumference on shaft divided by the number of active coils per inch of spring.

22.6.3 Example. A close wound extension spring made from 0.050 in. dia wire, with .40 in. OD, 232 active coils, 3.4 in. inside dia of ring, 25 lb IT, with half hooks joined to form a ring is expanded over a 6 in. dia shaft. What is the pressure per inch of circumference on the shaft?

$$FL = \pi(3.4 + .40) = 11.95 \text{ in.}$$

$$FL \text{ also} = d(TC + 1) + 2\,(ID/2)$$

$$= .050\,(232 + 1) + 2\,(.300/2)$$

$$= 11.65 + .30 = 11.95 \text{ in.}$$

$$F = \pi(6 + .40) - 11.95$$

$$= 20.10 - 11.95 = 8.15 \text{ in.}$$

$$\text{Rate} = r = \frac{G\ d^4}{8\ N\ D^3} = \frac{11.2 \times 10^6 \times .050^4}{8 \times 232.2 \times .350^3}$$

$$= .877 \text{ lb/in.}$$

(.2 coil allowed for deflection of 2 half hooks)

Pressure per inch of circumference on shaft,

$$\text{equals } 2\pi(.877) - \frac{2\pi 3.4 \times .877}{6} + \frac{2 \times 2.5}{6}$$

$$= 5.52 - 3.13 + .83 = 3.22 \text{ lb.}$$

23. TORSION SPRINGS

23.1 DESIGN FORMULAS. The stress in helical torsion springs is a bending or tensile stress. The stress caused by a load should be compared with the elastic limit in tension of the material to determine the allowable stress. Comparison should also be made with the curves of allowable stresses (corrected for torsion springs) as shown in Figure 62, Section II, of this Appendix. In Table VII for helical torsion springs two formulas are listed for each property. Either may be used; one is based on load P, the other on deflection F°, and the results should be the same.

23.2 STRESSES IN TORSION SPRING ENDS. Frequently the limiting stress value in helical torsion springs is the stress value in the ends. When a helical torsion spring has an eye as in Figure 46, or bent off the coil as in Figure 47 the stress at the inside of the bend is a tensile stress. The sharp curvature causes the neutral axis to move inward toward the center of the curve and the

TABLE VII. *Formulas for helical torsion springs*

Property	Round wire	Square wire	Rectangular wire *
Torque, lb in. T (also, PR)	$\frac{E\ d^4\ F^\circ}{4{,}000\ N\ D}$	$\frac{E\ d^4\ F^\circ}{2{,}375\ N\ D}$	$\frac{E\ b\ t^3\ F^\circ}{2{,}375\ N\ D}$
	$\frac{S_b\ d^3}{10.2}$	$\frac{S_b\ d^3}{6}$	$\frac{S_b\ b\ t^2}{6}$
Bending stress, psi S_b	$\frac{10.2\ P\ R}{d^3}$	$\frac{6\ P\ R}{d^3}$	$\frac{6\ P\ R}{b\ t^2}$
	$\frac{E\ d\ F^\circ}{392\ N\ D}$	$\frac{E\ d\ F^\circ}{392\ N\ D}$	$\frac{E\ t\ F^\circ}{392\ N\ D}$
Deflection, F°	$\frac{4{,}000\ P\ R\ N\ D}{E\ d^4}$	$\frac{2{,}375\ P\ R\ N\ D}{E\ d^4}$	$\frac{2{,}375\ P\ R\ N\ D}{E\ b\ t^3}$
	$\frac{392\ S_b\ N\ D}{E\ d}$	$\frac{392\ S_b\ N\ D}{E\ d}$	$\frac{392\ S_b\ N\ D}{E\ t}$
Change in moment $T_2 - T_1$	$\frac{F^\circ_2 - F^\circ_1}{\frac{F^\circ}{T}}$	$\frac{F^\circ_2 - F^\circ_1}{\frac{F^\circ}{T}}$	$\frac{F^\circ_2 - F^\circ_1}{\frac{F^\circ}{T}}$
ID after deflection in. ID_1	$\frac{N\ (ID\ free)}{N + \frac{F^\circ}{360}}$	$\frac{N\ (ID\ free)}{N + \frac{F^\circ}{360}}$	$\frac{N\ (ID\ free)}{N + \frac{F^\circ}{360}}$
Rate r_t lb. in./Deg	$\frac{T}{F^\circ}$	$\frac{T}{F^\circ}$	$\frac{T}{F^\circ}$

When a spring has (makes) several complete revolutions, F° = 360° multiplied by the number of revolutions.

* Rectangular wire may be coiled on edge or on flat, but b is always parallel to the axis of the spring and t is always perpendicular to the axis.

tensile stress becomes that of a cantilever loading multiplied by a constant (K). The formula for determining the stress in the bend of the eye in Figure 46 follows:

$$S_b = \frac{32\ P\ R\ K}{\pi\ d^3}$$

Where: K = Curvature Stress Correction Factor Figure 51

R = Mean Radius of Eye in.

$$= \frac{\text{ID of eye} + d}{2}$$

$$= \frac{\text{OD of eye} - d}{2}$$

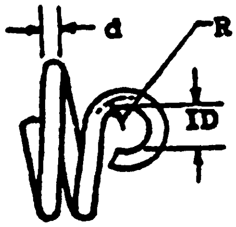


FIGURE 46

For bends off the coil as in Figure 47 the stress value in the bend is:

$$S_b = \frac{32\ P\ l_1\ K}{\pi\ d^3}$$

Where l_1 = distance from center of bend to load

K = stress correction factor Figure 51.

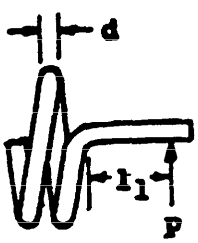


FIGURE 47

For the determination of K from Figure 51 in this instance D = 2 times the inside radius of the bend.

23.3 DEFLECTION OF TORSION SPRING ENDS. When the length of the material in the arms of a helical torsion spring approaches the length of material in one coil, the deflection of the arms will cause the deflection under applied loads to be in error (See Figure 48.) Such ends deflect as a cantilever and may be calculated as such or the formula for spring rate including arms may be used.

The formula for spring rate when the deflection of the arms should be included is:

$$r_t = \frac{Ed^4}{1170\left(L + \frac{l_1}{3} + \frac{l_2}{3}\right)}$$

l_1 = Length of arms from the center of the coil to the point of load, in.

l_2 = Length of arm from the center of the coil to the point of load, in.

r_t = Spring rate, lb in./degree.

L = Active length of material equals πD N

In springs with a large number of coils and short arms the deflection of thearms is neglected. However, short arms should be avoided as this causes difficulty in coiling and forming.

23.4 CHANGE IN DIAMETER AND LENGTH. When a helical torsion spring is deflected a reduction in diameter and an increase in length occurs. In order to prevent binding or scuffing, which reduces spring life, sufficient space must be provided when operating over a rod or in a cylinder. The new inside diameter ID_1 in a helical torsion spring due to deflection is obtained by the formula shown in Table VII. The shaft diameter should be slightly less than the calculated diameter to prevent binding and distortion in service. The change in length is due to the increase in the number of coils at the deflected position. If a helical torsion

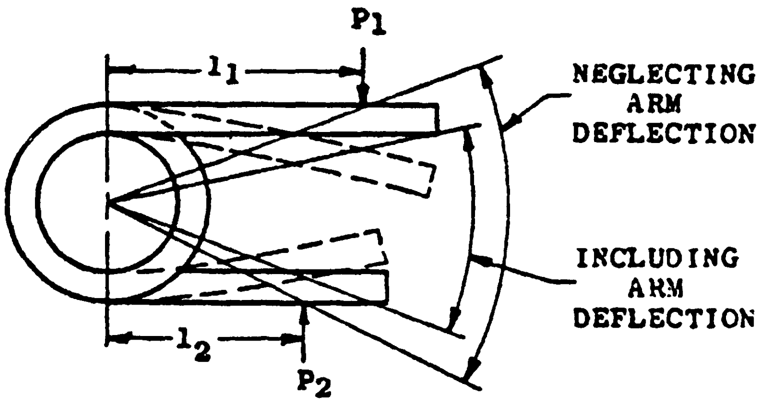


FIGURE 48

spring makes one complete revolution the increase in length is equal to one thickness of wire, plus an allowance for the space between coils, if any.

23.5 HELIX OF TORSION SPRINGS. The hand or direction of coiling (helix) should always be specified for torsion springs. A torsion spring should be so designed that the applied load tends to wind up the spring and increase its length. In springs operating under high stress it is desirable to design the springs with open coils. A slight space of about 1/64 inch or 20 to 25 per cent of the wire diameter will eliminate friction between coils and reduce stress concentration which will lengthen the spring life. When long helical torsion springs are used there exists the possibility of buckling. Since buckling will cause abrasion between coils, erratic loads and early spring failure, it should be avoided. Buckling may be reduced in varying amounts by providing some means of lateral support such as:

1. Mounting the spring over a rod or guide.
2. Mounting the spring in a tube.
3. Clamping the ends.
4. Winding the spring with a small amount of initial tension.

23.6 DESIGN NOMOGRAPHS FOR HELICAL TORSION SPRINGS. The nomographs in Figures 49 through 54 are for general guidance. They are based on a modulus E of 30,000,000 and can be used to reduce the time required to design a helical torsion spring. All results should be checked by the formulas in Table VII.

23.7 MOMENT VS. WIRE SIZE CHART. Table VIII is an aid to quickly determine the torque. (T or PR) that can be applied to a wire diameter at the suggested basic stress listed. For example, what wire diameter is required to support a torque of 10.5 in. lbs? From the table it will be observed that .090 in. diameter music wire or corrosion resisting steel; 0915 in diameter carbon or alloy steel and .125 in. diameter copper and nickel alloys (phosphor-bronze or monel) could be considered. The final determination should be arrived at by formula and evaluation of allowable stresses depending upon type of service. The basic stress indicated is a bending stress S_b caused by a torque T or PR, corrected for curvature.

FIBER STRESS (Not Corrected for Curvature) vs. MOMENT
Torsion Springs

Low Range — Wire Diameter .015" to .150"

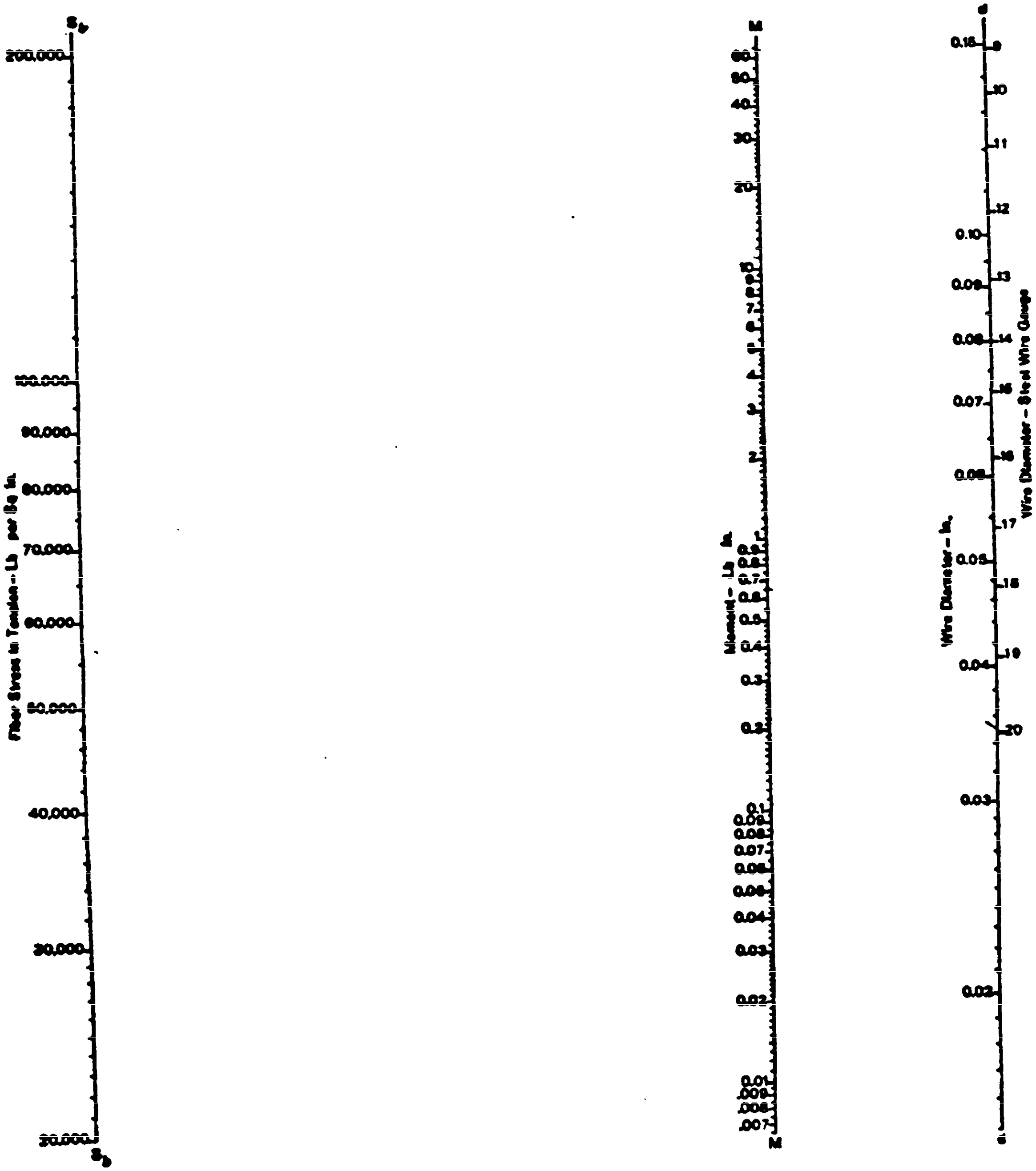


FIGURE 49

FIBER STRESS (Not Corrected for Curvature) vs. MOMENT
Torsion Springs

High Range — Wire Diameter .1" to 1"

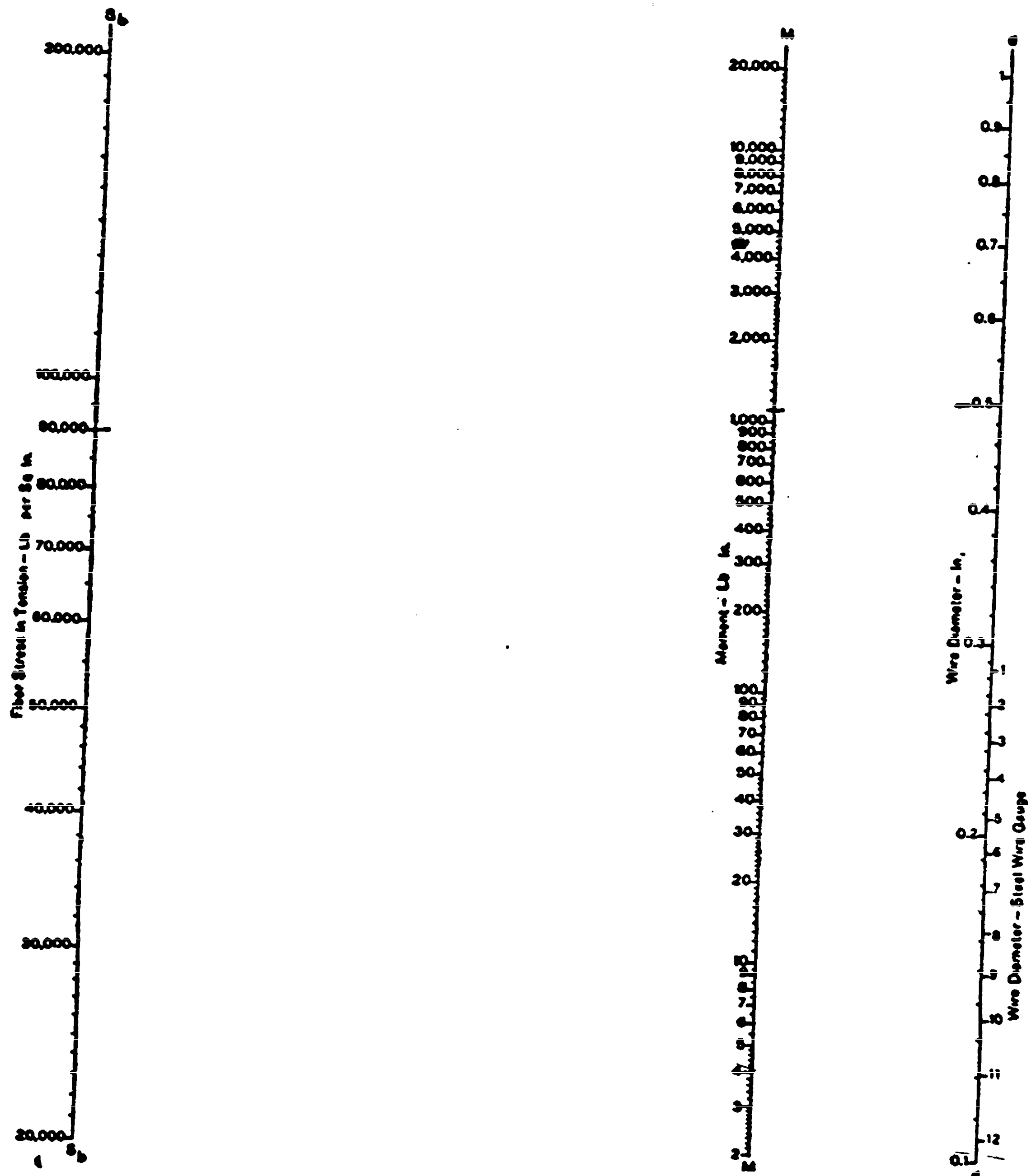


Figure 50

FIBER STRESS CORRECTION FOR CURVATURE
Torsion Springs

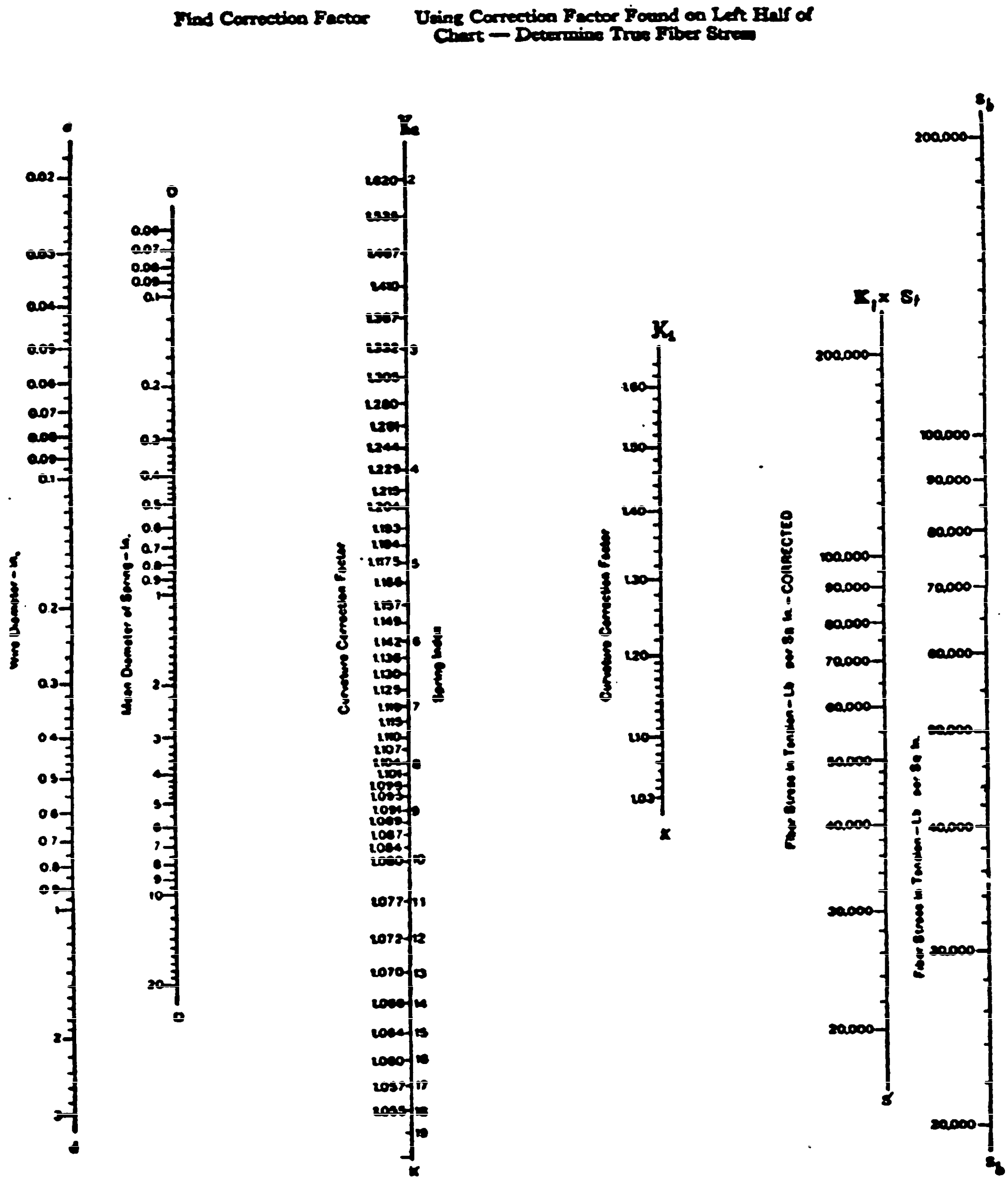


FIGURE 51

TURNS SPRING WILL GIVE PER COIL vs. MOMENT
Torsion Springs

Low Range {Mean Diameter: .1" to 3"
Wire Diameter: .015" to .150"

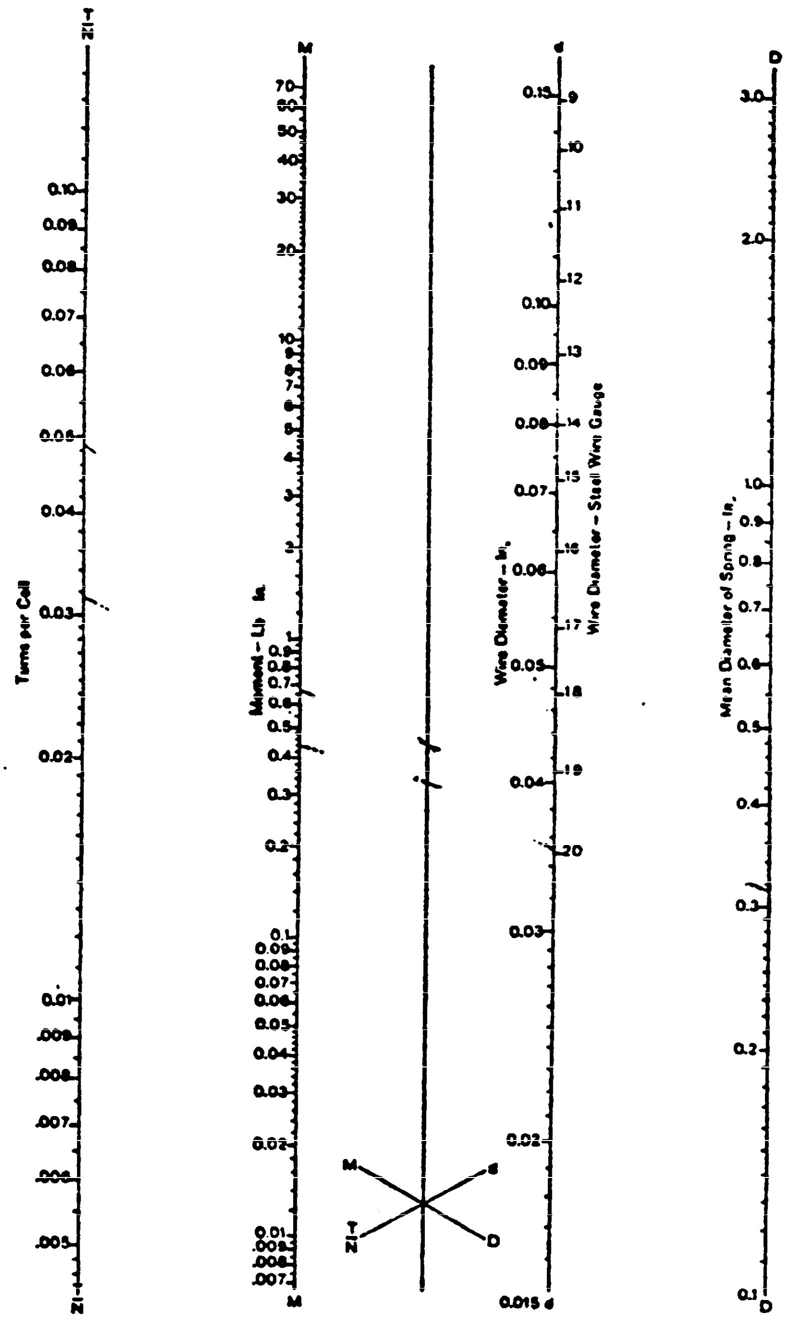


FIGURE 52

TURNS SPRING WILL GIVE PER COIL vs. MOMENT
Torsion Springs

High Range {Mean Diameter: 1" to 10"
Wire Diameter: .1" to 1"

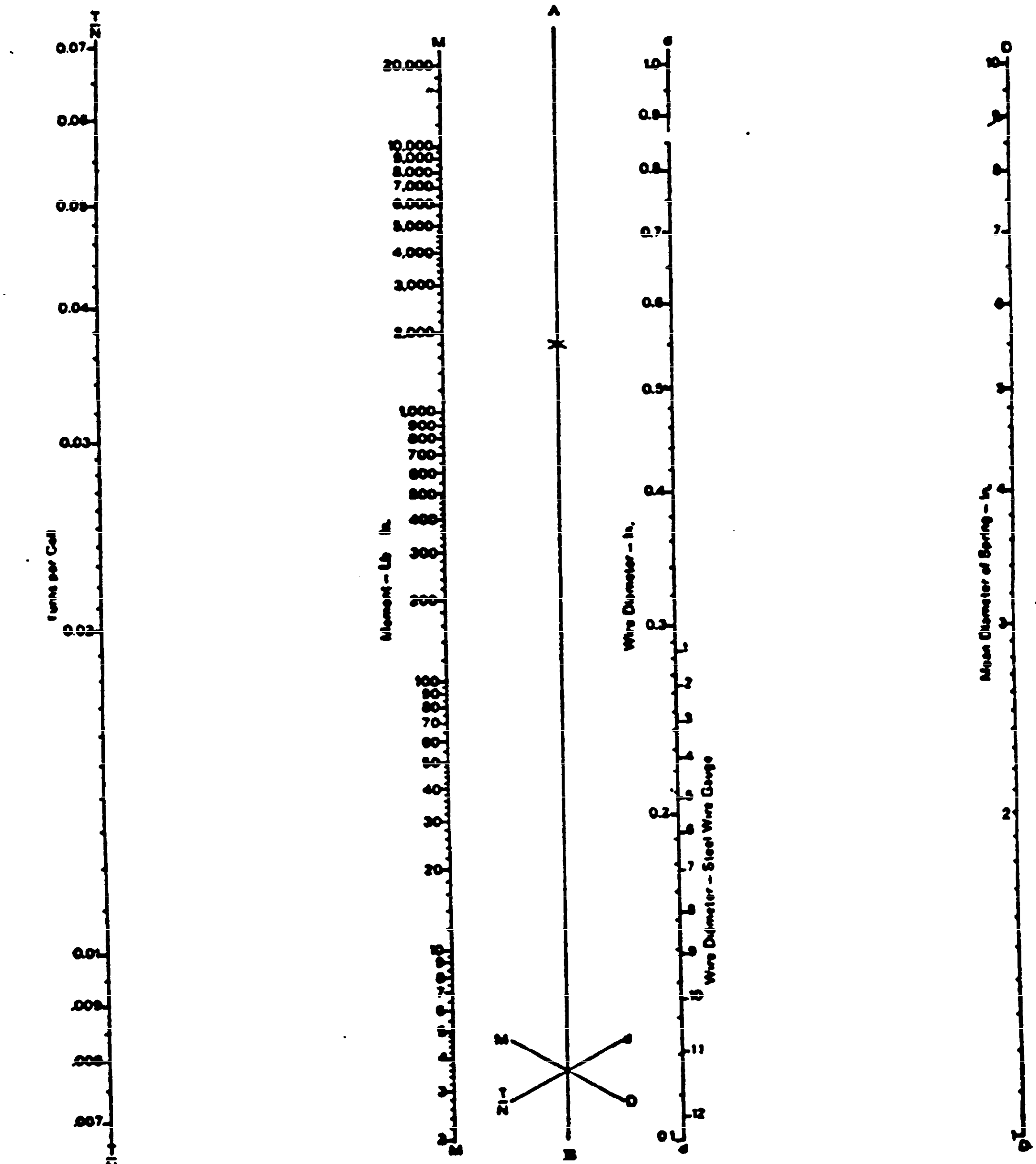


FIGURE 53

TURNS SPRING WILL GIVE PER COIL

Modulus (E) Other Than 30 × 10⁶

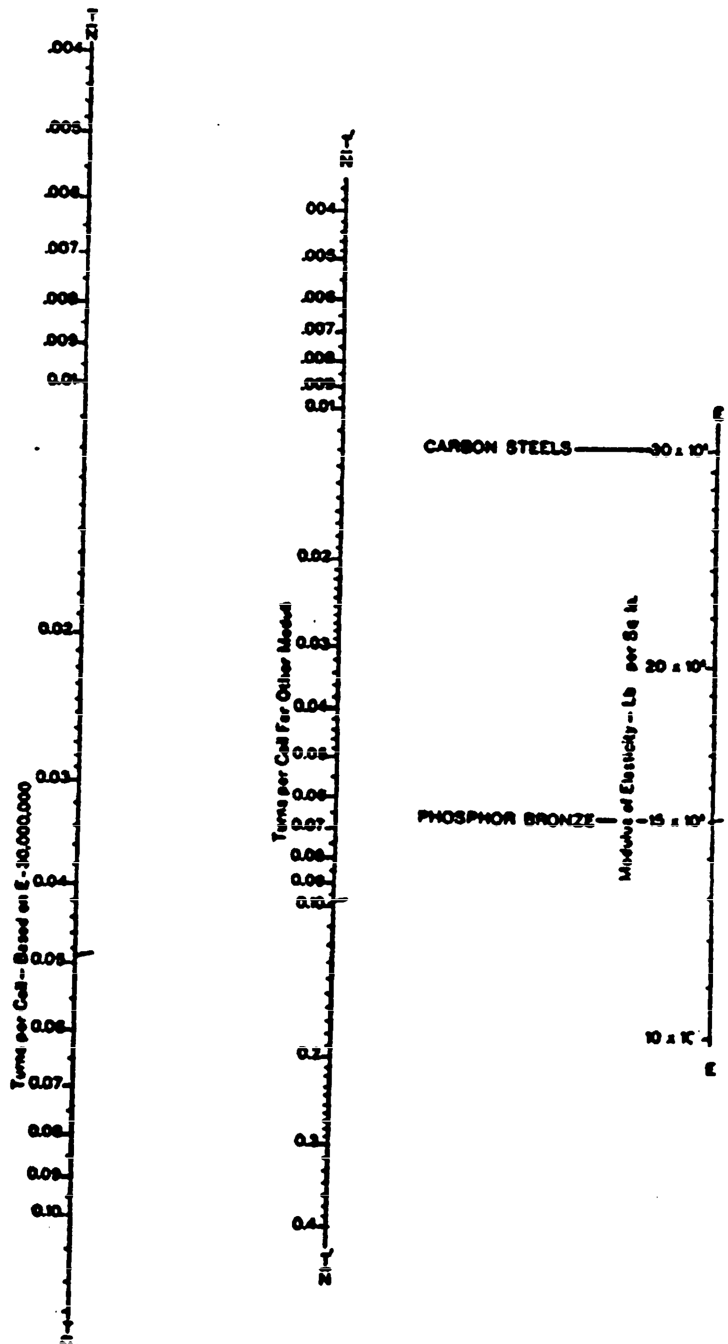


FIGURE 54

TABLE VIII. *Moment vs. wire size chart*

MUSIC WIRE			CARBON & ALLOY STEELS			COPPER & NICKEL ALLOYS		
Corrected Moment lb.-in.	Wire Diam. in.	Basic Stress psi	Corrected Moment lb.-in.	Wire Diam. in.	Basic Stress psi	Corrected Moment lb.-in.	Wire Diam. in.	Basic Stress psi
.0101	.008	201,000	1.013	.041	149,500	.00358	.008	71,200
.0143	.009	200,000	1.555	.0475	148,000	.00509	.009	71,000
.0196	.010	199,500	2.27	.054	146,700	.00694	.010	70,800
.0259	.011	198,500	3.47	.0625	145,000	.00925	.011	70,600
.0337	.012	198,000	5.24	.072	143,000	.0152	.013	70,500
.0425	.013	196,800	7.12	.080	141,500	.0189	.014	70,200
.0527	.014	195,500	10.49	.0915	139,800	.0281	.016	69,800
.0780	.016	194,000	11.21	.0937	139,000	.0398	.018	69,500
.1103	.018	193,000	15.72	.1055	136,400	.0546	.020	69,300
.1505	.020	192,000	23.05	.1205	134,000	.0827	.022	69,100
.199	.022	190,000	25.5	.125	133,000	.1056	.025	69,000
.256	.024	189,000	31.8	.135	131,700	.1635	.029	68,200
.323	.026	187,500	41.4	.1483	129,200	.219	.032	67,900
.443	.029	183,000	48.0	.1562	128,100	.310	.036	67,700
.533	.031	182,000	53.1	.162	127,100	.422	.040	67,200
.638	.033	181,000	68.0	.177	124,000	.598	.045	66,900
.755	.035	179,500	79.8	.1875	123,100	.864	.051	66,500
.886	.037	178,000	85.0	.192	122,400	1.194	.057	65,600
1.030	.039	177,000	104.6	.207	120,200	1.664	.064	64,600
1.185	.041	175,500	116.5	.2187	118,400	2.35	.072	64,100
1.350	.043	173,000	131.0	.2253	117,000	3.17	.080	63,200
1.535	.045	171,500	163	.2437	115,000	4.61	.091	62,300
1.73	.047	170,000	175	.250	114,000	6.43	.102	61,600
1.95	.049	169,000	199	.2625	112,200	8.82	.114	60,600
2.18	.051	167,500	239	.2812	109,600	11.5	.125	59,800
2.70	.055	165,000	301	.3065	106,400	12.2	.128	59,500
3.26	.059	162,000	315	.3125	105,300	17.2	.144	58,500
3.95	.063	161,000	367	.331	103,000	23.9	.162	57,200
4.70	.067	159,500	405	.3437	101,700	33.4	.182	56,200
5.50	.071	157,000	464	.3625	99,000	45.9	.204	55,100
6.39	.075	154,000	506	.375	97,800	63.5	.229	53,800
7.68	.080	153,000	575	.3938	95,800	88.7	.258	52,600
9.04	.085	150,000	623	.4062	94,600	122	.289	51,300
10.56	.090	147,500	753	.4375	91,800	169	.325	50,100
12.30	.095	146,500	898	.4687	88,800	234	.365	48,800
14.35	.100	146,000	1060	.500	86,500	320	.410	47,300
16.90	.106	145,000	1445	.5625	82,700	441	.460	46,200
19.30	.112	143,600	1910	.625	79,800			

NOTE: The values for Music Wire may also be used for Corrosion Resisting Steels.

23.8 HELICAL TORSION SPRING CALCULATION. The stress in a helical torsion spring is normally a bending stress (tension) and, for this reason E is used in the formulas. Tables and nomographs of characteristics for helical torsion springs aid in design. The wire diameter also can be obtained by solving an equation as in the following example.

23.8.1 Example. A torsion spring made of corrosion resisting steel Type FS 302 is required to exert a load of 9 lb at the end of a 2 in. arm of the spring (measuring from the center line of the spring to the point of contact) at 100 degrees of deflection; the ID being 1¼ in. Select a suitable wire diameter for average service and determine the number of coils, body length, etc. (assuming that the stress for average service from Figure 62 for torsion springs may be equal to 120,000 psi max)

Theoretical wire diameter d,

$$d = \sqrt[3]{\frac{10.2\ PR}{S_b}} = \sqrt[3]{\frac{10.2 \times 9 \times 2}{120{,}000}} = \sqrt[3]{0.0015} = 0.116 \text{ in.}$$

From Table I, select .120 inch diameter wire. Stress at the 9 lb load,

$$S_b = \frac{10.2\ PR}{d^3} = \frac{10.2 \times 9 \times 2}{.120^3}$$
$$= 106{,}300 \text{ psi}$$

From Figure 51, $K_1 = 1.075$ S max.
$= 106{,}300 \times 1.075 = 114{,}300$ max. psi

Since Figure 62 allows maximum working stress of 123,000 psi (82,000 × 1.5), wire diameter selected is satisfactory.

To determine a safe maximum deflection without permanent set beyond the final position, first find the maximum safe load.

From above we know that 9 lb will stress the material to 114,300 psi

From Figure 62, the minimum elastic limit for compression springs = 102,000 psi. Multiplying this by 1.5 = 153,000 psi, the stress for a safe maximum deflection without permanent set. The load to produce this stress is

$$\frac{9}{X} = \frac{114{,}300}{153{,}000}; X = \frac{9 \times 153{,}000}{114{,}300}$$
$= 12.04$ lb (say 12 lb)

Since spring rate $= \dfrac{T}{F^\circ} = \dfrac{9 \times 2}{100^\circ}$

.18 lb in./Deg, the additional deflection to produce a load of

12 lb is $\dfrac{12-9}{.18} = \dfrac{3.0}{.18}$

$= 16.7^\circ$

Number of active coils,

$$N = \frac{Ed^4\ F^\circ}{4{,}000\ PRD} = \frac{28{,}000{,}000 \times 0.120^4 \times 100}{4{,}000 \times 9 \times 2 \times 1.370}$$
$= 5.88$ (say, 6 coils)

Free length over coils,

$(6 + 1) \times 0.120 = 0.840$ in.

It is usually desirable to coil torsion springs with a slight space between the coils equal to about 20 to 25 per cent of the wire diameter. Assuming 20 per cent, the space equals $0.20 \times 0.120 = 0.024$ in. Six coils would have 6 spaces and equals $6 \times 0.024 = 0.144$ in.

The free length over coils would equal $0.840 + 0.144 = 0.984$ in. (say, 1.0 in.)

The ID of the spring reduces slightly due to deflection. After maximum deflection without permanent set has taken place,

$$ID_1 = \frac{N\,(\text{ID free})}{N + \dfrac{F^\circ}{360}} = \frac{6 \times 1.25}{6 + 115/360} = \frac{7.50}{6 + 0.32} = 1.18 \text{ in.}$$

The shaft over which such a spring is fitted should, therefore, be less than 1.18 in. A shaft $1\frac{5}{32}$ in. diameter would be satisfactory.

The letters P (lb load) and R (moment arm) may be replaced by T where T equals the torque in inch pounds. In this example, T would equal 9 × 2 = 18 in. lb.

Specifying loads in inch-pounds torque (T) or as pounds times the lever arm P×R gives the same results; either method may be used.

23.9 PRECAUTIONS AND SUGGESTIONS FOR EFFECTIVE DESIGN OF HELICAL TORSION SPRINGS.

(a) Always try to support a torsion spring by a rod running through the center of the spring. Torsion springs unsupported or held by clamps or lugs alone are unsteady, will buckle, and cause additional stresses in the wire.

(b) Torsion springs should be designed and installed so that the deflection increases the number of coils. This increase should be allowed for in the design of space requirements.

(c) The inside diameter reduces during deflection and should be computed to determine the clearance over the supporting rod.

(d) Use as few bends in the ends as possible. They are often formed in separate operations, are expensive, and cause concentrations of stress and frequent breakage.

(e) Consider tolerances on diameters when determining clearances over rods.

(f) Always specify the direction of coiling as either right-hand or left-hand on drawings. Right-hand coiling follows the same direction as standard bolt and screw threads.

(g) Springs may be closely or loosely wound, but they should not be wound tightly except when frictional resistance between the coils is desired.

(h) Avoid using double-torsion springs. Two single-torsion springs, one coiled left-hand and the other right-hand, usually can perform the same action as a double-torsion spring, at less than half the cost.

(i) When deflected $1\frac{1}{4}$ times the maximum deflection as assembled, the total stress should be less than the Minimum Elastic Limit shown by the curves in Figure 62, as modified by their multiplying constants.

24. SPIRAL TORSION SPRINGS

24.1 GENERAL. A spiral torsion spring delivers torque at its inner end to the shaft on which it is mounted and to the part on which its outer end is fastened.

24.2 STRESS. The stress is in bending and should be compared with the elastic limit in tension of the material to determine the allowable stress. The recommended allowable stress for thickness under .060 inch is 175,000 psi and for heavier sizes 150,000 psi for commercial spring steels. Lower stresses will increase fatigue life.

TABLE IX. *Formulas for spiral torsion springs of rectangular section*

Property	Formula
Bending Stress, psi S_b	$S_b = \dfrac{6PR}{bt^2}$
	$S_b = \dfrac{EtF^\circ}{114.6L}$
Torque, in. lb T	$T = P \times R$
	$T = \dfrac{S_b bt^2}{6}$
Active Length, in. L	$L = \dfrac{EtF^\circ}{114.6 S_b}$
	$L = \dfrac{\pi EUt}{S_b}$
Deflection F°	$F^\circ = \dfrac{114.6 S_b L}{Et}$
Rate r_t in. lbs/Deg	$\dfrac{T}{F^\circ}$

TABLE X. *Torque for 1 in. wide spiral torsion springs stressed at 100,000 psi (Other widths directly by proportion)*

t in.	T in. lbs	t in.	T in. lbs	t in.	T in. lbs	t in.	T in. lbs
.008	1.07	.025	10.4	.063	66.2	.105	182
.010	1.66	.032	17.1	.072	86.5	.125	260
.015	3.75	.041	28.1	.080	107.0	.156	410
.020	6.68	.054	48.7	.092	141.0	.188	588

24.3 ENDS. A variety of end shapes can be used. Inner ends usually are bent to fit in a slot. Outer ends usually are formed to fit over a bolt or pin.

25. CONSTANT FORCE SPRINGS

25.1 GENERAL. These springs have an appearance similar to clock or motor springs but are wound so that a constant force P causes a continuous unwinding of the coils. The springs are made from a strip of flat spring material which has been given a curvature by continuous heavy forming so that in its relaxed condition it is in the form of a tightly wound spiral. In Figure 56 the outer end has been extended by a constant force P and each incremental part of the straightened portion L has been deflected from its natural curvature in passing through the working zone X. The force P at any extension is determined only by the work required to straighten the material in zone X from its natural curvature in its coiled condition. The constant force giving a zero rate or gradient can be changed during manufacture to make springs with a slightly altered rate; thereby producing springs with a small negative, positive or changing rate if desired, but the zero rate has the broadest practical application possibilities. Such springs should have widths equal to 5 to 200 times (250 max) their thickness to have good stability, (100 times thickness is often used). They are not generally used in elevated temperatures over 140°F. Most applications are for the extension type, but these springs also may be used as motor springs to turn rollers at a constant torque.

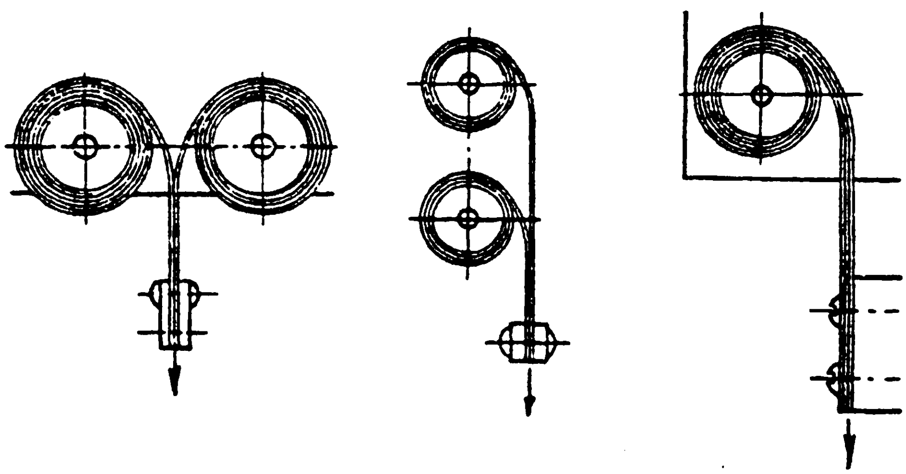


FIGURE 55

25.2 MATERIALS. The two most generally used spring materials and method of specifying them on drawings follow: STEEL, SPRING, TEMPERED, BLUE, SAE 1095, NO. 1 EDGE, HARDNESS RANGE Rc 48 to 51 and STEEL, CORROSION RESISTING AISI 302 ¾ HARD. Other materials are used if essential to design requirements. In specifying the length any additional material for forming special loops or straight portions should be added to the length determined by the formulas. If in a formula for thickness t, the result derived is an odd size such as .0096 inch, the next standard larger thickness such as .010 inch should be used.

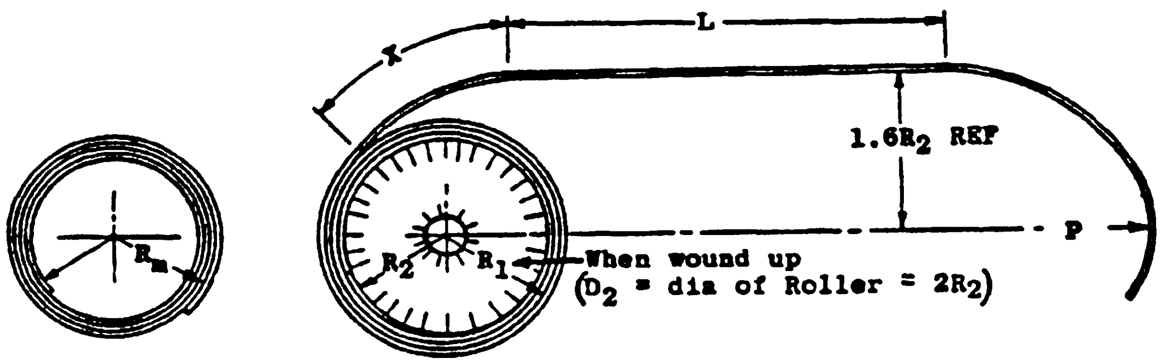
25.3 INNER ENDS. Normally the inner end of a constant force extension spring is held to the roller by its natural gripping action, when about 1½ turns of material remain on the roller at full deflection. No other fastening is required, except where there is some tendency for the material to be wound onto the roller in improper alignment or if danger of overtravel exists. In such cases the inner end may be bent and inserted into a slot in the roller, or retained by a small upset hook, or held by a small screw. Inner ends held by screws should be recessed and applied in a manner to prevent the screw head from deforming the natural curvature of the spring. Flanges on rollers also aid in guiding the flow of the material onto the rollers.

25.4 OUTER ENDS. Outer ends usually have round or pear shaped holes to fit over a screw. These ends may be left square, rounded or trimmed to suit. The ends also may be annealed and bent to form a loop and then riveted, for mounting over pins. The outer ends normally follow the regular curvature of the spring diameter, but they may, if desired, be manufactured with a preformed straight end to facilitate attachment.

25.5 MULTIPLE MOUNTING. Two springs

TABLE XI. *Stress factor S_f for varying fatigue life*

Material	Fatigue life					
	Up to 5,000	10,000	12,000	20,000	25,000	100,000 to 1,000,000
High carbon steel SAE 1095	.020	.0155	.014	.012	.010	.009
Corrosion resisting steels	.024	.019	.017	.015	.012	.009



FREE POSITION
UNMOUNTED

OPERATING POSITION
MOUNTED ON ROLLER

FIGURE 56

may be mounted back to back, each on its own roller, and thus double the load. Two springs also may be mounted in tandem, one above the other, to double the load. Two or more springs can also be mounted on the same roller (laminar mounting) to obtain multiple loads, see Figure 55.

25.6 **STRESS FACTOR.** When designing constant force springs, a stress factor, S_t, based on fatigue life, endurance limit, and actual tests, is used in place of the customary stress formulas because of the combination of stresses occurring in this type of spring. Values of S_t, determined for certain materials and fatigue life are given in Table XI. Intermediate values may be interpolated.

25.7 **EXTENSION TYPE.** The most commonly used is the extension type and the spring is supplied in a coil. It must then be rewound onto a roller which should be about 20 per cent larger in diameter than the inside diameter of the spring. An extension spring of this type is shown in Figure 56.

25.8 **FORMULAS.** The following formulas may be used to compute approximate proportions of the spring:

$$bt = \frac{26.4\ p}{E\ S_t^2}$$

10 Revolutions or Less

$$R_n = \sqrt{\frac{E\ b\ t^3}{26.4\ p}}$$

$$R_2 = 1.20\ R_n$$

Over 10 Revolutions

$$R_m = \sqrt{\frac{E\ b\ t^3}{26.4\ p}}$$

$$R_n = \frac{R_m}{1.20}$$

$$R_2 = 1.20\ R_m$$

$$L = F + 10\ R_2 \quad \text{or} \quad L = F + 5\ D_2$$

The following formula may be used to compute the adjusted load P which the spring will exert based on the proportions established by the formulas above:

$$P = \frac{E\ b\ t^3}{26.4}\left[\frac{1}{R_n^2} - \left(\frac{1}{R_n} - \frac{1}{R_1}\right)^2\right]$$

The abbreviations in 20.1.3 are used plus the following:

R_n = natural radius of curvature in inches (minimum) in the free position = ID/2.

R_m = natural radius of curvature in inches (minimum) in the free position = ½ (OD — 2t).

R_1 = radius of outer coil in inches when mounted. It is the expanded radius of curvature due to material build up = ½ (OD — 2t) where OD equals the roller diameter plus allowance for the number of coils used.

R_2 = radius of roller over which spring is mounted.

D_2 = Diameter of roller = 2 R_2.

25.9 **SIMPLIFIED DESIGN.** The design of extension type constant force springs can be simplified by replacing the modulus of elasticity E, and other terms by a varying factor Q. The values of Q have been worked out for springs having 10 coils or less and the simplified formula for load P follows:

$$P = Q\ b\ t$$

TABLE XII. *Factor Q for varying fatigue life*

Material	Fatigue life								
	2,500	4,000	5,000	10,000	15,000	20,000	35,000	100,000	1,000,000
High carbon steel SAE 1095	....	521	417.5	270.5	....	169	123	101.2	81.3
Corrosion resisting steels	660	....	502	....	316	233	151	86.9	69.4

CONSTANT FORCE SPRING DESIGN CHART HIGH CARBON STEELS

CHART A — FATIGUE LIFE 4,000 CYCLES MINIMUM

Q=521

t	R2	b	1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1¼	1½	2	2½	3	4
			LOAD P														
.002	.174	.207	.130	.195	.260	.325	.390										
.003	.261	.313	.185	.278	.370	.463	.555	.740	.925								
.004	.348	.418		.390	.520	.650	.780	1.04	1.30	1.56	2.08						
.005	.435	.523			.650	.813	.975	1.30	1.63	1.95	2.60						
.006	.522	.627			.780	.975	1.17	1.56	1.95	2.34	3.12	3.90					
.007	.609	.730				1.14	1.36	1.82	2.28	2.73	3.64	4.55	5.46				
.008	.696	.835					1.56	2.08	2.60	3.12	4.16	5.20	6.24				
.010	.870	1.04					1.95	2.60	3.25	3.90	5.20	6.50	7.80	10.4			
.012	1.04	1.25						3.12	3.90	4.68	6.24	7.80	9.36	12.5			
.014	1.22	1.46							4.55	5.46	7.28	9.10	10.9	14.6	18.2		
.015	1.31	1.57								5.85	7.80	9.75	11.7	15.6	19.5	23.4	
.016	1.39	1.67									8.32	10.4	12.5	16.6	20.8	25.0	
.018	1.57	1.88									9.36	11.7	14.0	18.7	23.4	28.1	
.020	1.74	2.09									10.4	13.0	15.6	20.8	26.0	31.2	41.6
.022	1.91	2.27										14.3	17.2	22.9	28.6	34.3	45.8
.025	2.18	2.62											19.5	26.0	32.5	39.0	52.0
.028	2.44	2.92											21.8	29.1	36.4	43.7	58.3
.032	2.78	3.34												33.3	41.6	49.9	66.6

CHART B — FATIGUE LIFE 5,000 CYCLES MINIMUM

Q=417.5

t	R2	b	1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1¼	1½	2	2½	3	4
			LOAD P														
.002	.200	.240	.104	.156	.208	.260	.312										
.003	.300	.360	.156	.234	.312	.390	.469	.623	.782								
.004	.400	.480		.312	.415	.520	.623	.830	1.04	1.25	1.66						
.005	.500	.600			.520	.650	.780	1.04	1.30	1.56	2.08						
.006	.600	.720			.623	.780	.936	1.25	1.56	1.87	2.50	3.12					
.007	.700	.840				.910	1.09	1.46	1.82	2.18	2.91	3.64	4.37				
.008	.800	.960					1.25	1.67	2.08	2.50	3.33	4.16	5.00				
.010	1.00	1.20					1.56	2.08	2.60	3.12	4.16	5.20	6.24	8.32			
.012	1.20	1.44						2.50	3.12	3.74	5.00	6.24	7.49	9.98			
.014	1.40	1.68							3.64	4.36	5.82	7.28	8.73	11.6	14.5		
.015	1.50	1.80								4.68	6.24	7.80	9.36	12.5	15.6	18.7	
.016	1.60	1.92									6.66	8.32	9.99	13.3	16.6	20.0	
.018	1.80	2.16									7.50	9.36	11.2	15.0	18.7	22.4	
.020	2.00	2.40									8.32	10.4	12.5	16.6	20.9	24.9	33.3
.022	2.20	2.64										11.4	13.7	18.3	22.9	27.4	36.6
.025	2.50	3.00											15.6	20.8	26.0	31.2	41.6
.028	2.80	3.36											17.4	23.3	29.0	34.8	46.4
.032	3.20	3.84												26.6	33.3	39.7	53.2

CHART C — FATIGUE LIFE 10,000 CYCLES MINIMUM

Q=270.5

t	R2	b	1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1¼	1½	2	2½	3	4
			LOADS P														
.002	.260	.312	.067	.101	.135	.169	.202										
.003	.390	.468	.101	.152	.202	.253	.304	.405	.506								
.004	.520	.625		.202	.270	.337	.405	.539	.675	.811	1.08						
.005	.649	.779			.337	.422	.506	.675	.845	1.01	1.35						
.006	.779	.935			.405	.506	.607	.801	1.01	1.22	1.62	2.03					
.007	.910	1.09				.592	.708	.945	1.17	1.42	1.89	2.36	2.84				
.008	1.04	1.25					.810	1.08	1.35	1.62	2.16	2.70	3.24				
.010	1.30	1.56					1.01	1.35	1.69	2.03	2.70	3.37	4.05	5.40			
.012	1.56	1.87						1.62	2.02	2.43	3.24	4.05	4.85	6.48			
.014	1.82	2.19							2.37	2.84	3.78	4.73	5.67	7.56	9.45		
.015	1.95	2.34								3.04	4.04	5.05	6.06	8.10	10.1	12.1	
.016	2.08	2.50									4.32	5.37	6.46	8.64	10.8	12.9	
.018	2.34	2.81									4.86	6.06	7.29	9.70	12.1	14.5	
.020	2.60	3.12									5.44	6.75	8.10	10.8	13.5	16.2	21.6
.022	2.85	3.42										7.40	8.93	11.9	14.8	17.8	23.7
.025	3.25	3.90											10.1	13.5	16.9	20.2	27.0
.028	3.64	4.36											11.3	15.1	18.9	22.7	30.2
.032	4.15	4.98												17.8	22.2	26.7	35.6

CHART D — FATIGUE LIFE 20,000 CYCLES MINIMUM

Q=169

t	R2	b	1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1¼	1½	2	2½	3	4
			LOAD P														
.002	.336	.403	.042	.063	.084	.105	.126										
.003	.504	.605	.063	.095	.126	.158	.189	.252	.315								
.004	.672	.806		.126	.168	.210	.252	.336	.420	.504	.672						
.005	.840	1.01			.210	.263	.315	.420	.525	.630	.840						
.006	1.01	1.21			.252	.315	.378	.504	.630	.756	1.01	1.26					
.007	1.18	1.42				.367	.441	.587	.735	.882	1.18	1.47	1.76				
.008	1.34	1.61					.504	.673	.840	1.01	1.34	1.68	2.02				
.010	1.68	2.02					.631	.840	1.05	1.26	1.68	2.10	2.52	3.36			
.012	2.02	2.42						1.01	1.26	1.51	2.02	2.52	3.02	4.03			
.014	2.35	2.82							1.47	1.76	2.35	2.94	3.53	4.70	5.88		
.015	2.52	3.02								1.89	2.52	3.15	3.78	5.04	6.30	7.56	
.016	2.69	3.23									2.69	3.36	4.03	5.37	6.72	8.06	
.018	3.02	3.63									3.02	3.78	4.54	6.05	7.56	9.07	
.020	3.36	4.03									3.36	4.20	5.04	6.72	8.40	10.1	13.4
.022	3.70	4.44										4.62	5.54	7.39	9.24	11.1	14.8
.025	4.20	5.04											6.30	8.40	10.5	12.6	16.8
.028	4.71	5.64											7.06	9.41	11.8	14.1	18.8
.032	5.37	6.44												10.8	13.4	16.1	21.5

FIGURE 57

CONSTANT FORCE SPRING
DESIGN CHART
HIGH CARBON STEELS

CHART E — FATIGUE LIFE 35,000 CYCLES MINIMUM

Q=123

t	2R	D_1	b→ 1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1 1/4	1 1/2	2	2 1/2	3	4
			LOAD P														
.002	.400	.480	.0301	.0452	.0604	.0767	.0903										
.003	.600	.720	.0462	.0693	.0921	.115	.138	.183	.231								
.004	.800	.960		.0921	.123	.154	.185	.246	.307	.369	.493						
.005	1.00	1.20			.154	.192	.231	.308	.384	.461	.616						
.006	1.20	1.44			.184	.231	.277	.369	.461	.554	.738	.923					
.007	1.40	1.68				.261	.323	.431	.538	.646	.862	1.07	1.29				
.008	1.60	1.92					.369	.492	.615	.738	.984	1.23	1.48				
.010	2.00	2.40					.461	.615	.769	.923	1.23	1.54	1.85	2.46			
.012	2.40	2.88						.738	.923	1.11	1.48	1.84	2.21	2.95			
.014	2.80	3.36							1.07	1.29	1.72	2.15	2.58	3.44	4.31		
.015	3.00	3.60								1.38	1.85	2.31	2.77	3.69	4.61	5.54	
.016	3.20	3.84									1.97	2.46	2.95	3.94	4.92	5.90	
.018	3.60	4.32									2.22	2.77	3.32	4.43	5.54	6.64	
.020	4.00	4.80									2.46	3.08	3.69	4.92	6.15	7.38	9.84
.022	4.40	5.28										3.38	4.06	5.41	6.77	8.12	10.8
.025	5.00	6.00											4.61	6.15	7.69	9.23	12.3
.028	5.60	6.72											5.16	6.89	8.61	10.3	13.8
.032	6.40	7.68												7.88	9.84	11.8	15.7

CHART F — FATIGUE LIFE 100,000 CYCLES MINIMUM

Q=101.2

t	2R	D_1	b→ 1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1 1/4	1 1/2	2	2 1/2	3	4
			LOAD P														
.002	.444	.533	.0253	.0379	.0506	.0632	.0758										
.003	.667	.800	.0379	.0569	.0759	.0949	.114	.152	.189								
.004	.889	1.07		.0758	.101	.126	.152	.202	.253	.303	.404						
.005	1.11	1.33			.126	.158	.189	.253	.316	.379	.505						
.006	1.33	1.60			.152	.189	.227	.303	.379	.455	.606	.758					
.007	1.56	1.87				.221	.265	.354	.441	.530	.707	.884	1.06				
.008	1.78	2.14					.303	.404	.505	.606	.808	1.01	1.21				
.010	2.22	2.66					.379	.505	.631	.758	1.01	1.26	1.52	2.02			
.012	2.67	3.20						.606	.758	.909	1.21	1.52	1.82	2.42			
.014	3.11	3.73							.884	1.06	1.41	1.77	2.12	2.83	3.54		
.015	3.33	4.00								1.14	1.51	1.89	2.27	3.03	3.79	4.55	
.016	3.56	4.27									1.62	2.02	2.42	3.23	4.04	4.85	
.018	4.00	4.80									1.82	2.27	2.73	3.64	4.55	5.45	
.020	4.44	5.33									2.02	2.53	3.03	4.04	5.05	6.07	8.08
.022	4.89	5.87										2.78	3.33	4.44	5.56	6.67	8.89
.025	5.55	6.67											3.79	5.05	6.31	7.58	10.1
.028	6.22	7.46											4.24	5.73	7.07	8.48	11.3
.032	7.11	8.53												6.54	8.08	9.70	12.9

CHART G — FATIGUE LIFE 1,000,000 CYCLES MINIMUM

Q=81.3

t	2R	D_1	b→ 1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1 1/4	1 1/2	2	2 1/2	3	4
			LOAD P														
.002	.500	.600	.0203	.0305	.0407	.0508	.0610										
.003	.750	.900	.0305	.0457	.0610	.0762	.0914	.122	.152								
.004	1.00	1.20		.0610	.0813	.102	.122	.163	.203	.244	.325						
.005	1.25	1.50			.102	.127	.152	.203	.254	.305	.407						
.006	1.50	1.80			.122	.152	.183	.244	.305	.366	.488	.609					
.007	1.75	2.10				.178	.213	.285	.356	.427	.569	.711	.854				
.008	2.00	2.40					.244	.325	.406	.488	.650	.812	.975				
.010	2.50	3.00					.305	.407	.508	.610	.813	1.02	1.22	1.63			
.012	3.00	3.60						.488	.610	.732	.976	1.22	1.46	1.95			
.014	3.50	4.20							.711	.854	1.14	1.42	1.71	2.28	2.84		
.015	3.75	4.50								.915	1.22	1.52	1.83	2.43	3.04	3.66	
.016	4.00	4.80									1.30	1.63	1.95	2.60	3.25	3.90	
.018	4.50	5.40									1.46	1.83	2.19	2.93	3.66	4.39	
.020	5.00	6.00									1.63	2.03	2.44	3.25	4.06	4.88	6.50
.022	5.50	6.60										2.24	2.68	3.58	4.47	5.37	7.15
.025	6.25	7.50											3.05	4.07	5.08	6.10	8.13
.028	7.00	8.40											3.41	4.55	5.69	6.83	9.11
.032	8.00	9.60												5.20	6.50	7.80	10.4

FIGURE 58

CONSTANT FORCE SPRING
DESIGN CHART
CORROSION RESISTANT STEEL

CHART H — FATIGUE LIFE 2,500 CYCLES MINIMUM

LOAD P (Q=660)

t	2R.	D.	b→ 1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1¼	1½	2	2½	3	4
.002	.145	.174	.164	.246	.329	.411	.492										
.003	.218	.261	.247	.370	.493	.616	.740	.988	1.32								
.004	.291	.349		.493	.658	.822	.986	1.32	1.64	1.97	2.63						
.005	.363	.436			.821	1.03	1.23	1.64	2.05	2.46	3.28						
.006	.436	.523			.986	1.23	1.48	1.97	2.47	2.96	3.95	4.94					
.008	.581	.697					1.97	2.63	3.29	3.95	5.24	6.58	7.90				
.010	.727	.873					2.48	3.30	4.12	4.95	6.60	8.25	9.90	13.2			
.012	.873	1.05						3.96	4.95	5.94	7.92	9.90	11.9	15.8			
.014	1.02	1.22							5.77	6.93	9.24	11.5	13.9	18.5	23.1		
.016	1.16	1.40							6.60	7.92	10.6	13.2	15.8	21.1	26.4	31.7	
.018	1.31	1.57							7.44	8.89	11.9	14.9	17.8	23.8	29.7	35.5	
.020	1.46	1.75							8.25	9.90	13.2	16.5	19.8	26.4	33.0	39.6	52.8
.022																	
.025	1.82	2.18									16.5	20.6	24.8	33.0	41.2	49.5	66.0
.028																	
.031	2.26	2.71												40.9	51.1	61.3	81.0

CHART J — FATIGUE LIFE 5,000 CYCLES MINIMUM

(Q=502)

t	2R.	D.	b→ 1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1¼	1½	2	2½	3	4
.002	.171	.205	.126	.183	.253	.315	.379										
.003	.257	.308	.189	.283	.377	.473	.567	.757	.946								
.004	.343	.412		.379	.505	.630	.757	1.01	1.26	1.51	2.01						
.005	.428	.513			.632	.789	.947	1.26	1.57	1.88	2.51						
.006	.514	.617			.757	.945	1.13	1.51	1.88	2.27	3.02	3.76					
.008	.684	.821					1.51	2.01	2.51	3.02	4.02	5.06	6.06				
.010	.855	1.03					1.98	2.51	3.14	3.77	5.02	6.28	7.53	10.0			
.012	1.03	1.23						3.01	3.77	4.52	6.03	7.53	9.04	12.1			
.014	1.20	1.44							4.39	5.27	7.03	8.79	10.5	14.1	17.6		
.016	1.37	1.64							5.01	6.03	8.04	10.0	12.1	16.1	20.1	24.1	
.018	1.54	1.85							5.65	6.80	9.04	11.3	13.6	18.1	22.6	27.1	
.020	1.71	2.05							6.28	7.53	10.0	12.6	15.1	20.1	25.1	30.1	40.2
.022																	
.025	2.14	2.56									12.6	15.7	18.8	25.1	31.4	37.7	50.2
.028																	
.031	2.65	3.18												31.1	38.9	46.7	62.4

CHART K — FATIGUE LIFE 13,000 CYCLES MINIMUM

(Q=316)

t	2R.	D.	b→ 1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1¼	1½	2	2½	3	4
.002	.222	.263	.079	.119	.158	.198	.237										
.003	.333	.400	.119	.178	.237	.296	.356	.474	.593								
.004	.444	.533		.236	.316	.395	.474	.632	.790	.948	1.26						
.005	.555	.665			.395	.494	.592	.790	.988	1.19	1.58						
.006	.666	.798			.474	.593	.711	.948	1.19	1.42	1.90	2.37					
.008	.888	1.06					.948	1.26	1.58	1.90	2.52	3.16	3.79				
.010	1.11	1.33					1.19	1.58	1.98	2.37	3.16	3.95	4.74	6.33			
.012	1.33	1.59						1.90	2.37	2.84	3.79	4.74	5.69	7.58	9.48		
.015	1.66	1.99								3.56	4.74	5.93	7.11	9.48	11.7	14.2	
.018	2.00	2.40									5.69	7.11	8.53	11.4	14.2	17.1	
.020	2.22	2.65									6.32	7.90	9.48	12.6	15.8	19.0	25.3
.022	2.42	2.91										8.69	10.4	13.9	17.4	20.9	27.8
.025	2.78	3.33											11.9	15.8	19.8	23.7	31.6
.028	3.11	3.73												17.7	22.1	26.5	35.4
.031	3.44	4.12												19.6	24.5	29.4	39.2

CHART L — FATIGUE LIFE 20,000 CYCLES MINIMUM

LOAD P (Q=233)

t	2R.	D.	b→ 1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1¼	1½	2	2½	3	4
.002	.263	.316	.0583	.0874	.117	.145	.174										
.003	.395	.474	.0874	.131	.175	.218	.262	.350	.437								
.004	.526	.631		.175	.233	.291	.350	.466	.582	.699	.932						
.005	.656	.787			.291	.364	.437	.583	.728	.874	1.17						
.006	.789	.947			.350	.437	.524	.699	.874	1.05	1.40	1.75					
.008	1.05	1.26					.699	.932	1.16	1.40	1.86	2.33	2.80				
.010	1.32	1.58					.874	1.17	1.46	1.75	2.33	2.91	3.50	4.66			
.012	1.58	1.90						1.40	1.75	2.09	2.80	3.50	4.19	5.59	6.99		
.015	1.98	2.37								2.62	3.50	4.37	5.24	6.99	8.74	10.5	
.018	2.37	2.84									4.19	5.24	6.29	8.39	10.5	12.6	
.020	2.63	3.16									4.66	5.83	6.99	9.32	11.7	14.0	18.6
.022	2.89	3.47										6.41	7.69	10.3	12.8	15.4	20.5
.025	3.30	3.96											8.74	11.7	14.6	17.5	23.3
.028	3.68	4.42												13.0	16.3	19.6	26.1
.031	4.08	4.89												14.4	18.1	21.7	28.9

FIGURE 59

CONSTANT FORCE SPRING
DESIGN CHART
CORROSION RESISTANT STEEL

CHART M — FATIGUE LIFE 35,000 CYCLES MINIMUM

t	$2R_n$	D_o / b →	1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1¼	1½	2	2½	3	4
			LOAD P														
.002	.331	.397	.0390	.0570	.0760	.0950	.114										
.003	.496	.595	.0570	.0855	.114	.143	.171	.228	.285								
.004	.662	.794		.114	.152	.190	.228	.304	.380	.456	.608						
.005	.827	.992			.190	.238	.285	.380	.475	.570	.760						
.006	.991	1.19			.228	.285	.342	.456	.570	.684	.912	1.14					
.008	1.32	1.58					.456	.608	.760	.912	1.22	1.52	1.82				
.010	1.65	1.98					.570	.760	.950	1.14	1.52	1.90	2.28	3.04			
.012	1.98	2.38						.912	1.14	1.37	1.82	2.28	2.74	3.65	4.56		
.015	2.48	2.98								1.71	2.28	2.85	3.42	4.56	5.70	6.84	
.018	2.98	3.58									2.74	3.42	4.10	5.47	6.84	8.21	
.020	3.31	3.97									3.04	3.80	4.56	6.08	7.60	9.12	12.2
.022	3.64	4.37										4.18	5.02	6.69	8.36	10.0	13.4
.025	4.14	4.97											5.70	7.60	9.50	11.4	15.2
.028	4.63	5.56												8.51	10.6	12.8	17.0
.031	5.03	6.04												9.42	11.8	14.1	18.9

Q=151

CHART N — FATIGUE LIFE 100,000 CYCLES MINIMUM

t	$2R_n$	D_o / b →	1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1¼	1½	2	2½	3	4
			LOAD P														
.002	.445	.534	.0217	.0326	.0435	.0543	.0652										
.003	.666	.800	.0326	.0489	.0652	.0815	.0978	.130	.163								
.004	.890	1.07		.0652	.0869	.109	.130	.174	.217	.261	.348						
.005	1.11	1.33			.109	.136	.163	.217	.272	.326	.435						
.006	1.33	1.59			.130	.163	.196	.261	.326	.391	.521	.652					
.008	1.78	2.14					.261	.348	.434	.521	.695	.869	1.04				
.010	2.22	2.66					.326	.435	.543	.652	.869	1.09	1.30	1.74			
.012	2.67	3.20						.522	.652	.782	1.04	1.30	1.56	2.09	2.61		
.015	3.33	4.00								.978	1.30	1.63	1.96	2.61	3.26	3.91	
.018	4.00	4.80									1.56	1.95	2.35	3.13	3.91	4.69	
.020	4.44	5.33									1.74	2.17	2.61	3.48	4.35	5.21	6.95
.022	4.89	5.87										2.39	2.87	3.82	4.78	5.74	7.64
.025	5.55	6.66											3.26	4.35	5.43	6.52	8.69
.028	6.13	7.46												4.87	6.08	7.30	9.73
.031	6.90	8.28												5.39	6.74	8.08	10.8

Q=86.9

CHART P — FATIGUE LIFE 1,000,000 CYCLES MINIMUM

t	$2R_n$	D_o / b →	1/8	3/16	1/4	5/16	3/8	1/2	5/8	3/4	1	1¼	1½	2	2½	3	4
			LOAD P														
.002	.500	.600	.0174	.0260	.0347	.0434	.0521										
.003	.750	.900	.0260	.0390	.0521	.0651	.0781	.104	.130								
.004	1.00	1.20		.0520	.0694	.0868	.104	.139	.174	.208	.278						
.005	1.25	1.50			.0868	.108	.130	.174	.217	.260	.347						
.006	1.50	1.80			.104	.130	.156	.208	.260	.312	.416	.52					
.008	2.00	2.40					.208	.278	.347	.416	.555	.694	.832				
.010	2.50	3.00					.260	.347	.434	.521	.694	.868	1.04	1.39			
.012	3.00	3.60						.416	.521	.625	.833	1.04	1.25	1.67	2.08		
.015	3.75	4.50								.781	1.04	1.30	1.56	2.08	2.60	3.12	
.018	4.50	5.40									1.25	1.56	1.87	2.50	3.12	3.75	
.020	5.00	6.00									1.39	1.74	2.08	2.78	3.47	4.16	5.55
.022	5.50	6.60										1.91	2.29	3.05	3.81	4.58	6.11
.025	6.25	7.50											2.60	3.47	4.34	5.21	6.94
.028	7.00	8.40												3.89	4.86	5.83	7.77
.031	7.75	9.30												4.30	5.38	6.45	8.61

Q=69.4

FIGURE 60

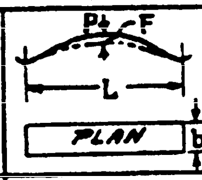
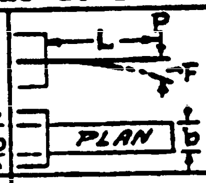
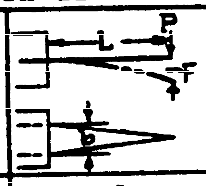
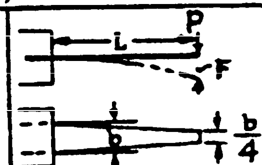
In using the simplified formula, b and t ratios can be estimated. For length L the regular formulas apply. See Figures 57 to 60 as these values apply and were determined from the simplified formula with values of Q as shown in Table XII.

25.10 TABLES OF SPRING CHARACTERISTICS (DESIGN CHARTS) FOR EXTENSION TYPE CONSTANT FORCE SPRINGS. As as aid to the selection of constant force springs, combinations of width b, thickness t, natural radius of curvature R_n, roller diameter D_2 (20 per cent larger than ID of spring). for high carbon spring steel SAE 1095 and corrosion resisting steels are shown in Figuers 57 to 60. Lengths should be determined by the regular formula $L=F + 10\ R_n$ or $L = F + 5\ D_2$.

25.11 EXAMPLE. If a high carbon steel extension type constant force spring is required to deflect 24 inches, have a fatigue life of 4000 cycles and exert a load P of 10.4 pounds, it could be made, as shown in Figure 57 Chart A, with a width b of 1 inch, thickness t of .020 inch, provided that the roller diameter D_2 equals 2.09 inches and the inside diameter

TABLE XIII

FORMULAS FOR FLAT SPRINGS
(Based upon standard beam formulas where the deflection is small)

PROPERTY				
Deflection F Inches	$\frac{P\ L^3}{4\ E\ b\ t^3}$	$\frac{4\ P\ L^3}{E\ b\ t^3}$	$\frac{6\ P\ L^3}{E\ b\ t^3}$	$\frac{5.22\ P\ L^3}{E\ b\ t^3}$
	$\frac{S_b\ L^2}{6\ E\ t}$	$\frac{2\ S_b\ L^2}{3\ E\ t}$	$\frac{S_b\ L^2}{E\ t}$	$\frac{.87\ S_b\ L^2}{E\ t}$
Load P Pounds	$\frac{2\ S_b\ b\ t^2}{3\ L}$	$\frac{S_b\ b\ t^2}{6\ L}$	$\frac{S_b\ b\ t^2}{6\ L}$	$\frac{S_b\ b\ t^2}{6\ L}$
	$\frac{4\ E\ b\ t^3\ F}{L^3}$	$\frac{E\ b\ t^3\ F}{4\ L^3}$	$\frac{E\ b\ t^3\ F}{6\ L^3}$	$\frac{E\ b\ t^3\ F}{5.22\ L^3}$
Stress S_b Bending psi	$\frac{3\ P\ L}{2\ b\ t^2}$	$\frac{6\ P\ L}{b\ t^2}$	$\frac{6\ P\ L}{b\ t^2}$	$\frac{6\ P\ L}{b\ t^2}$
	$\frac{6\ E\ t\ F}{L^2}$	$\frac{3\ E\ t\ F}{2\ L^2}$	$\frac{E\ t\ F}{L^2}$	$\frac{E\ t\ F}{.87\ L^2}$
Thickness t Inches	$\frac{S_b\ L^2}{6\ E\ F}$	$\frac{2\ S_b\ L^2}{3\ E\ F}$	$\frac{S_b\ L^2}{E\ F}$	$\frac{.87\ S_b\ L^2}{E\ F}$
	$\sqrt[3]{\frac{P\ L^3}{4\ E\ b\ F}}$	$\sqrt[3]{\frac{4\ P\ L^3}{E\ b\ F}}$	$\sqrt[3]{\frac{6\ P\ L^3}{E\ b\ F}}$	$\sqrt[3]{\frac{5.22\ P\ L^3}{E\ b\ F}}$

of the spring in the free position equals 1.74 inches.

The length L required for 24 inches of deflection F. is obtained from the formula $L = F + 5\,D_2$ and equals: $24 + 5\,(2.09) = 34.45$ inches (say 34½ in.).

To adjust an available load P in the charts to some other desired load P^1, select from the chart a value of P just greater than P^1 and determine R_a^1 by the following formula: (holding b and t constant).

$$R_a^1 = R_a \sqrt{\frac{P}{P^1}}$$

26. FLAT SPRINGS

26.1 GENERAL. Load requirements are intimately connected with spring dimensioning and the space available for the spring. The point of load application, deflection, length, width, and thickness should be clearly specified. Formulas in the following table may be used to determine various flat spring characteristics.

26.2 STRESS. The stress is in bending and should be compared with the elastic limit in tension of the material to determine the allowable stress. The recommended allowable stress for thickness under .060 inch is 175,000 psi and for heavier sizes 150,000 psi for commercial spring steels. Lower stresses will increase fatigue life.

27. CONED DISC (BELLEVILLE) SPRINGS

27.1 GENERAL. The coned disc (Belleville) spring or washer is a plain dished washer of a particular diameter, sectional profile, and height suited for an intended purpose. It is used in a variety of applications, all having

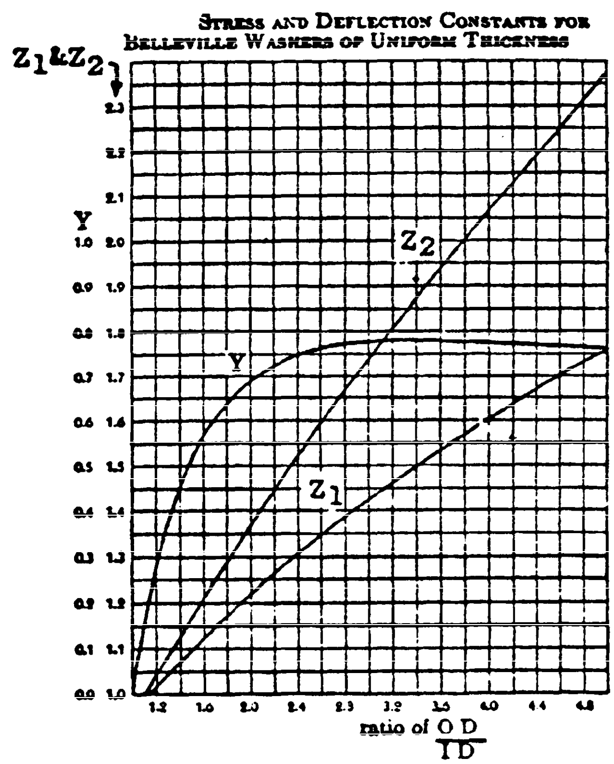


Figure 61

Recommended maximum working stresses for compression springs
(Fatigue Strength Curves)

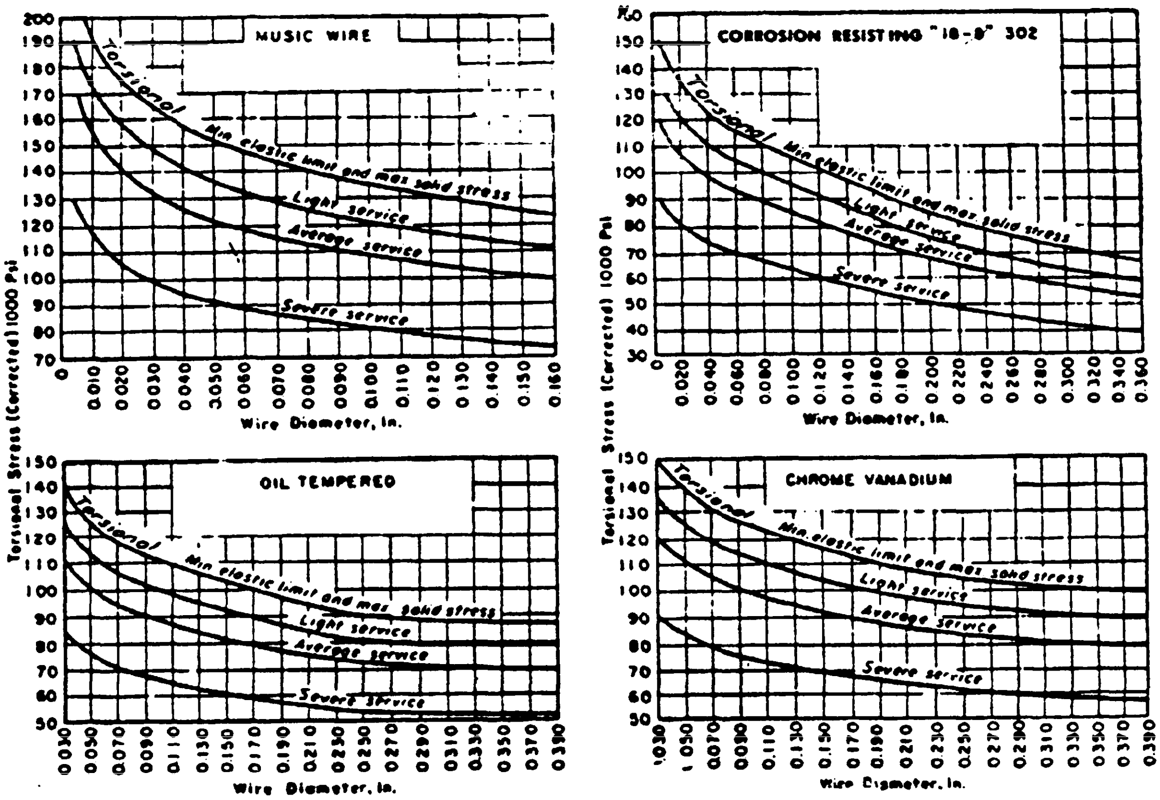


FIGURE 62
(Use with Table XIV)

the common characteristic of necessity for short range of motion and attendant high loads. In order to calculate the free spring height and required thickness of stock in a relatively simple manner, it is necessary to know the outside diameter (OD), inside diameter (ID). and the load (P) for a specific deflection.

27.2 LOAD DEFLECTION. By obtaining the value for constant (Y) from the proper curve (Figure 61) the following formula may be used to calculate load deflection characteristics.

$$P = \frac{E f}{(1-\sigma^2) Y a^2}\left[\left(h - \frac{f}{2}\right)(h-f)\, t + t^3\right]$$

Where:

a = one-half the outside diameter, in.

h = free height minus thickness, in.

27.3 STRESS. By obtaining the values for constants Z_1 and Z_2 from the proper curves (Figure 61) the fololwing formua may be used to calculate stress:

$$S_s = \frac{E f}{(1-\sigma^2)\, Y a^2}\left[Z_1\left(h - \frac{f}{2}\right) + Z_2\, t\right]$$

It is possible for the term $(h - f/2)$ to become negative if f is large. When this occurs, the terms inside the bracket should be changed to read

$$Z_1\,(h - f/2) - Z_2\, t$$

Recommended maximum working stresses for compressions springs—Continued
(Fatigue Strength Curves)

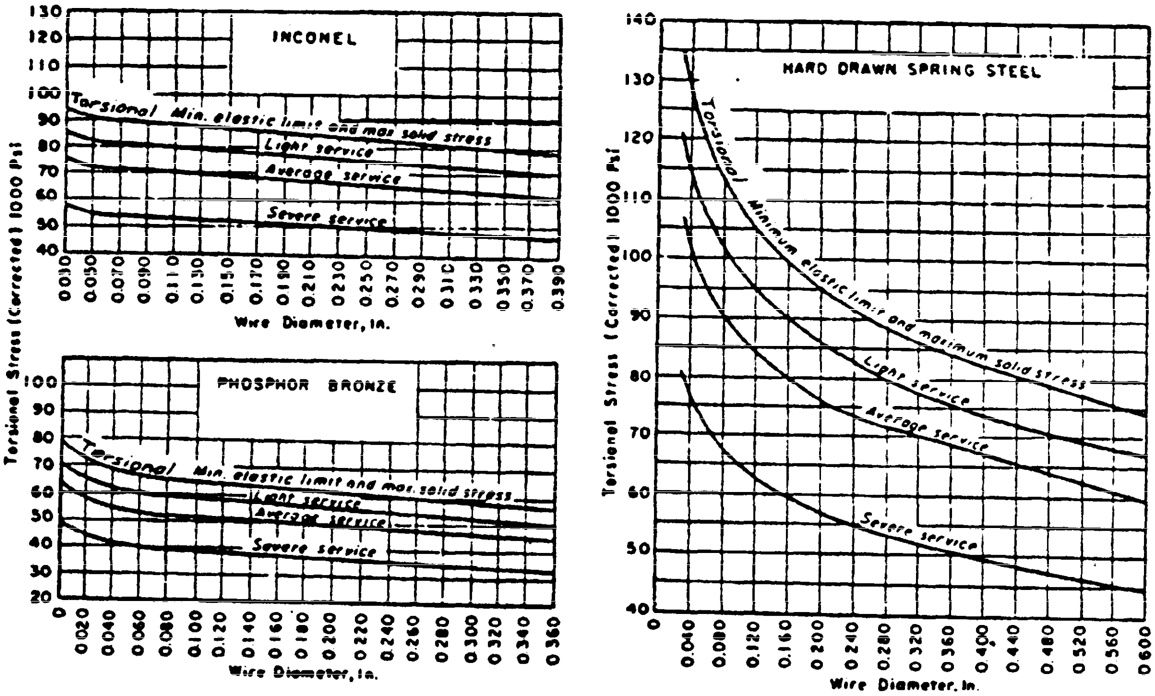


FIGURE 62—*Continued*
(Use with Table XIV)

This means that, in this instance, the maximum stress is a tensile stress. For a spring life of less than one-half million stress cycles a stress of 200,000 psi can be substituted for S_b, even though this might be slightly beyond the elastic limit of the steel. This is because the stress is calculated at the point of greatest intensity, which is on an extremely small part of the disk. Immediately surrounding this area is a much lower stressed portion which so supports the higher stressed point that very little settling results at atmospheric temperatures. For higher than atmospheric temperatures and long spring life, lower stresses must be employed.

27.4 LOADS. When five coned disc (Belleville) springs are stacked in series as in Figure 15, they will have a spring rate only one-fifth that of one disc and the solid load will be the same as for one disc.

When six discs are stacked in parallel as in Figure 15, they will have a spring rate and a solid load six times that of one disc, disregarding friction.

When six discs are stacked in parallel-series as in Figure 15, they will have a spring rate only two-thirds that of one disc and the solid load will be twice that of one disc, disregarding friction.

28. RECOMMENDED MAXIMUM WORKING STRESSES

28.1 FATIGUE STRENGTH CURVES. The fatigue strength curves Figure 62 are for the most popular spring materials. These are for compression springs, based on the minimum torsional elastic limit of each material. The values may be increased 25 per cent for springs that are properly stress relieved, cold set and shotpeened.

TABLE XIV
CRITICAL STRESS DATA
(For use in association with Figure 62.)*

COMPRESSION SPRING	EXTENSION SPRING	TORSION SPRING
1. TORSION STRESS—Compare calculated stress in coils with *service* curve of Fig. 62. 2. SOLID STRESS — Compare torsion stress in coils when compressed solid with *minimum elastic limit* curve.	1. TORSION STRESS (COILS) — Compare calculated design stress in coils with *service curve* of Fig. 62 multiplied by .85. 2. TORSION STRESS (HOOKS)— Compare calculated design stress in hooks with *service* curve multiplied by .85. 3. BENDING STRESS (HOOKS)— Compare calculated design stress in hooks with *service* curve multiplied by 1.5. 4. TORSION STRESS (COILS) AT MAX EXTENDED LENGTH— Compare calculated stress at this length with *min elastic limit* curve multiplied by 8.5. 5. TORSION STRESS (HOOKS) AT MAX EXTENDED LENGTH—Compare calculated stress at this length with *min elastic limit curve* multiplied by .85. 6. BENDING STRESS (HOOKS) AT MAX EXTENDED LENGTH—Compare calculated stress at this length with *min elastic limit* curve multiplied by 1.5.	1. BENDING STRESS (COILS) —Compare calculated design stress in coils with *service* curve of Fig. 62 multiplied by 1.5. 2. BENDING STRESS (ENDS) —Compare calculated design stress in ends with *service* curve multiplied by 1.5. 3. BENDING STRESS IN COILS AT MAXIMUM DEFLECTION — Compare calculated stress in coils at this deflection with *min elastic limit* of Fig. 62 multiplied by 1.5. 4. BENDING STRESS IN ENDS AT MAXIMUM DEFLECTION — Compare calculated stress in ends at this deflection with *min elastic limit* of Fig. 62 multiplied by 1.5.

* Note 1: After tentative spring configuration has been determined, use data in above table in association with Figure 62, to ascertain that allowable stresses are not exceeded.

Note 2: The above referenced "calculated design stresses" are TOTAL STRESSES. They include curvature stress-correction factors of Figures 39 and 51 (see para 21.11), except for extension spring hook stresses which include correction factor in basic formulas.

28.1.1 Light Service. This includes springs subjected to static loads or small deflections and seldom used springs such as those in bomb fuzes, projectiles, and safety devices. This service is for 1000 to 10,000 deflections.

28.1.2 Average Service. This includes springs in general use in machine tools, mechanical products and electrical components. Normal frequency of deflections not exceeding 3600 per hour permit such springs to withstand 100,000 to 1,000,000 deflections.

28.1.3 Severe Service. This includes springs subjected to rapid deflections over long periods of time and to shock loading such as in pneumatic hammers, hydraulic controls and valves. This service is for 1,000,000 deflections and above. Lowering the values 10 per cent permits 10,000,000 deflections.

28.1.4 Other Materials. For materials not shown on the curves in Figure 62, the following multiplying constants may be used.

(a) For Beryllium Copper, multiply the values of the Phosphor Bronze curves by 1.20.

(b) For Spring Brass, multiply the values of the Phosphor Bronze curves by .75.

(c) For Monel, multiply the values of the Inconel curves by .82.

(d) For K-Monel, multiply the values of the Inconel curves by .90.

(e) For Duranickel, use the same values as for Inconel.

(f) For Inconel X, (drawn to spring temper and precipitation hardened) multiply the values of the Inconel curves by 1.25.

(g) For Silico-Manganese, multiply the values of the Chrome-Vanadium curves by .90.

(h) For Chrome-Silicon, multiply the values of the Chrome-Vanadium curves by 1.20.

(i) For Valve Spring Quality Wire, use the same values as for Chrome-Vanadium.

(j) For Corrosion Resisting Steels type FS304 and FS420, multiply the values of the Corrosion Resisting Steel curves by .95.

(k) For Corrosion Resisting Steel type FS316, multiply the values of the Corrosion Resisting Steel curves by .90.

(l) For Corrosion Resisting Steels type AISI 431 and 17-7 PH, multiply the values of the Music Wire curves by .90.

28.2 PERMISSIBLE ELEVATED TEMPERATURES. Springs used at high temperatures exert less load and have larger deflections under load than at room temperature. Compression and extension springs subjected to the temperatures and stresses shown in the following table will have a loss of load of 5 per cent or less, (or if the load remains constant, they will deflect an additional 5 per cent), in 48 hours. Elastic limits and modulus values are also reduced, thus necessitating the lower allowable working stresses.

TABLE XV. *Permissible elevated temperatures for compression and extension springs. Loss of load at these temperatures is less than 5 percent in 48 hours.*

Spring material	Permissible elevated temperature F deg	Maximum recommended working stress St PSI
Brass Spring Wire	150	30,000
Phosphor Bronze	225	35,000
Music Wire	250	75,000
Beryllium-Copper	300	40,000
Hard Drawn Steel Wire	325	50,000
Carbon Spring Steels	375	55,000
Alloy Spring Steels	400	65,000
Monel	425	40,000
K-Monel	450	45,000
Duranickel	500	50,000
Corrosion Resisting FS-302	550	55,000
Corrosion Resisting AISI 431	600	50,000
Inconel	700	50,000
High Speed Steel	775	70,000
Cobenium, Elgiloy	800	75,000
Inconel X	850	55,000
Chrome-Moly-Vanadium	900	55,000

APPENDIX A, SECTION III

30. SPRING MANUFACTURE

30.1 GENERAL. Information on only a few operations is essential for design purposes.

1. STRESS RELIEVING. The usual types of hardening and tempering ovens are used for stress relieving. Springs made from prehardened wire such as Music Wire, Oil Tempered, Hard Drawn, Corrosion Resisting 18–8 and similar materials are stress relieved by heating at low temperatures from 400 to 650° F. to reduce the residual stresses trapped in the wire during the coiling operation. Springs made from annealed wire are hardened and tempered in a manner somewhat similar to tool steel. Precipitation hardening materials such as Beryllium-Copper, K-Monel, Inconel X, 17–7 PH and others are heated at varying temperatures depending upon composition, for extended times from 1 hour to 16 hours.

2. COLD SET TO SOLID. This process is used to stabilize the free length of a compression spring, so that subsequent inadvertent, or intentional, compression to solid height will not change the loads at working deflections.

(a) If a compression spring is designed so that the elastic limit is not exceeded when the spring is compressed to solid height, no appreciable permanent set will occur, other than removal of small kinks in the wire. The note "Cold Set to Solid" should be specified on the drawing of such springs.

(b) If a spring is designed so that the elastic limit is exceeded when the spring is closed solid, permanent set will occur and the free length will be decreased. Residual stresses of opposite sign will be set up in the wire when the load is released, so that if the spring again is closed solid it will withstand a higher calculated stress than the stress corresponding with the elastic limit. If the initial free length of the spring is made greater than the calculated free length by the proper amount, overstressing the spring beyond the elastic limit by compressing it to solid height will stabilize the characteristics and produce the desired loads at working deflections. Additional cycles of compressing the spring to solid height and releasing the load will not further change the free length. However, there is a limit to this process. After a certain initial free length has been reached for a particular spring, the final free length after compression to a solid height will remain constant no matter what increases are made in initial free length.

(c) When a spring is designed so that the stress at solid height is so far above the elastic limit that the spring will not have the desired loads at working deflections if cold set to solid, the note "Shall compress to .... in. without permanent set" should be placed on the drawing. The computed stress at the specified length (equal to, or less than, the final assembled length) must be less than the stress at the elastic limit. Whenever practicable, this design of spring should be discarded in favor of a spring having a solid stress within limits that will permit closing solid without permanent set.

Several methods may be used to cold set to solid. Small air presses and foot presses can be used for light springs, power presses, and specifically built motorized equipment for larger springs, and hydraulic presses for heavy springs. Light springs also may be compressed by hand over close fitting arbors.

33. GRINDING. End coils of compression springs are ground whenever it is necessary for the springs 1. to stand upright, 2. to obtain a good seat against a contacting part, 3. to reduce buckling, and 4. to cause the springs to exert more uniform pressures under a diaphragm or against a mating part. This operation is expensive and should be avoided wherever it is practicable to do so—especially on light springs with wire diameters under $\frac{1}{32}$ inch and where a large spring index ratio (OD/d) prevails, such as 13 or larger. Several types of disc grinding machines are available. The small disc grinders using paddles to hold springs for grinding are especially useful for small production runs from 100 to 5000 or more springs. Grinders with horizontal "ferris wheel' methods of loading are useful for larger quantities of light springs using wire diameters up to about $\frac{3}{32}$ inch. Heavier sizes may be ground in disc grinders with vertical ferris wheels.

34. SHOT PEENING. Spring life can be increased at least 30 per cent and has often been increased from two to ten times by shot peening. This process may be applied to all highly stressed springs made from steel and non-ferrous materials usually over $\frac{1}{16}$ inch wire diameter. Extension springs and closely wound torsion springs are difficult to shotpeend because the tiny steel shot is frequently trapped between the coils and is difficult to remove. The large increase in fatigue life of helical springs due to shot peening is accomplished by a combination of effects:

(a) Small surface irregularities seen only by the microscope are hammered smooth.

(b) The surface of the wire is thoroughly cleaned, and sharp burrs are made dull.

(c) This additional cold work hardens the surface of the wire and raises the physical properties where the stress is highest.

(d) Cold forging traps beneficial compression stresses near the wire surface which must be overcome by the destructive tensile stresses that cause fatique failure before breakage can occur. All heat-treating of springs and all stress-relieving processes should be done prior to shot peening, except in those instances where electroplating is used; it is then necessary to reheat after plating. Heating the springs above 500° F. after shot peening counteracts much of the beneficial effects of the trapped compression stresses produced by shot peening.

Springs made from annealed, oil tempered or alloy steels that must be electroplated can be shot peened principally to clean the surface, thus avoiding the necessity of soaking them in acid solutions to remove scale. The slightly roughened surface of shot peened springs does not produce a bright glossy, electroplated coating.

35. PROTECTIVE COATINGS. Uncoated or oil dipped springs are satisfactory where corrosive conditions are not a factor. Black japanning is often used as it is a flexible, inexpensive finish suitable for many applications. Enamels, lacquers and paint are occasionally used. Cadmium with supplementary chromate treatment provides one of the best electro deposited coatings because it is both flexible and corrosion resistant.

Nickel, chromium, zinc, and other metals are also deposited electrolytically. Black oxide and phosphate coatings are occasionally specified. Care must be used in applying coatings, first to use detergents, solvents, alkaline or vaporizing degreasers to remove oil and dirt, shotblasting or sand blasting to remove scale, and baking to remove the hydrogen embrittlement that may be caused by acid dipping or electroplating.

36. HYDROGEN EMBRITTLEMENT. Steel, particularly hardened steel, is susceptible to embrittlement resulting from hydrogen introduced by acid pickling, electroplating, or cathodic electrocleaning operations. Absorbed hydrogen results in brittle behavior particularly under sustained loading in the presence of stress concentrations. Baking to relieve hydrogen embrittled springs should be accomplished by a method similar to that specified in specification QQ–Z–325 or QQ–P–416.

(a) Most hydrogen is evolved during acid pickling and can be reduced by using organic inhibitors in the solution. The cleaning of springs by shotblasting or by tumbling is advisable so that less time is required in the pickling bath.

(b) Cathodic electrocleaning, by making the work anode instead of cathode, reduces the amount of hydrogen evolved and shortens the time required in acid dipping. This procedure is seldom used for springs because of the long time required to perform this operation for large quantities.

(c) Hydrogen evolution during electroplating can be reduced by using low current densities. Low voltages (between 6 and 8 volts) may be used. Improper or speedup methods are harmful. Commercial electroplating of springs may be unsatisfactory, due to the high current densities commercially used which cause a greater evolution of hydrogen and, therefore, more embrittlement.

(d) All electroplated springs made from steel should be dehydrated as soon as possible after plating. A time delay of 5 hours may prove too long in some cases.

High carbon spring steel is more susceptible to hydrogen embrittlement than the low-carbon steels, and faulty heat-treatment prior to plating will produce a type of grain structure that tends to increase embrittlement. Springs should not be plated while in a stressed condition or when part of an assembly, because stressed conditions increase embrittlement. For this reason the stretching of extension springs, so that deposition will occur between the coils, is not recommended.

Dehydrogenation, or removing of hydrogen, restores the normal condition of the steel. It is accomplished by heating the electroplated springs in an oven and should be done as soon as the springs are plated and washed. Correct temperatures and length of time at heat depend upon the type and size of the steel. Recommended heat for air circulating ovens is 375° ± 25° F for not less than 3 hours after plating.

Occasional embrittlement may occur even with the best regulated methods. Nonferrous

TABLE XVI. *Tolerances on outside diameter for all helical springs*

Outside diameter (in.)	Plus or minus tolerance on OD — Spring index (OD/d)*		
	Up to 6 (small)	6 to 10 (average)	10 to 16 (large)
1/8	0.002	0.003	0.004
1/8 to 3/16	0.003	0.004	0.006
3/16 to 1/4	0.004	0.006	0.008
1/4 to 3/8	0.006	0.008	0.010
3/8 to 1/2	0.008	0.010	0.012
1/2 to 3/4	0.010	0.012	0.015
3/4 to 1	0.012	0.015	0.022
1 to 1 1/2	0.015	0.022	0.031
1 1/2 to 2	0.022	0.031	0.047
2 to 3	0.047	0.062	0.094
3 to 4	0.062	0.094	0.125
4 to 5	0.094	0.125	0.156

* The spring index is based on OD for convenience.

TABLE XVII. *Tolerances on the number of coils for all helical springs*

Compression	Extension		Torsion
	Number of coils	Tolerance plus or minus	
In order to meet the requirements of load, rate, free length, solid height, it is necessary to vary the number of coils by plus or minus: 5 per cent	3 to 5	20°	Same as extension wherever the design permits. Otherwise the number of coils will vary in accordance with the position of the ends as shown in following table.
	6 to 8	30°	
	9 to 12	40°	
	For each additional coil add 1 1/2° per coil.		

Closer tolerances require trimming after coiling and increases manufacturing time and cost.

TABLE XVIII. *Tolerances on position of arms for torsion springs (For indexes OD/d up to 16)*

Total coils	Tolerance plus or minus
Up to 3	8°
Over 3 including 10	10°
Over 10 including 20	15°
Over 20 including 30	20°
Over 30	25°

TABLE XIX. *Tolerances on free length for compression springs with ends closed and ground and for accurately made extension springs*

Length spring, in.	Total number of coils	Plus or minus tolerance on length, in. — Spring index (O.D. ÷ d) — Up to 6 (small)	6 to 10 (average)	10 to 16 (large)
Up to ¾, inc.	Up to 4	0.007	*0.011*	0.015
	Over 4 to 7	0.011	0.015	0.018
	Over 7 to 11	0.015	0.018	0.022
	Over 11	0.018	0.022	0.026
Over ¾ to 1½, inc.	Up to 5	0.010	*0.015*	0.020
	Over 5 to 10	0.015	0.020	0.025
	Over 10 to 15	0.020	0.025	0.030
	Over 15	0.025	0.030	0.035
Over 1½ to 2½, inc.	Up to 10	0.020	*0.030*	0.040
	Over 10 to 20, inc.	0.030	0.040	0.050
	Over 20 to 30, inc.	0.040	0.050	0.060
	Over 30	0.050	0.060	0.070
Over 2½ to 3½, inc.	Up to 15	0.030	*0.045*	0.060
	Over 15 to 30	0.045	0.060	0.075
	Over 30 to 45	0.060	0.075	0.090
	Over 45	0.075	0.090	0.105
Over 3½ to 4½, inc.	Up to 20	0.040	*0.060*	0.080
	Over 20 to 40	0.060	0.080	0.100
	Over 40 to 60	0.080	0.100	0.120
	Over 60	0.100	0.120	0.140
Over 4½ to 6, inc.	Up to 25		*0.075*	0.100
	Over 25 to 50	0.075	0.100	0.125
	Over 50 to 75	0.100	0.125	0.150
	Over 75	0.125	0.150	0.175
Over 6 to 8, inc.	Up to 35	0.070	*0.105*	0.140
	Over 35 to 70	0.105	0.140	0.175
	Over 70 to 100	0.140	0.175	0.210
	Over 100	0.175	0.210	0.245
Over 8 to 10, inc.	Up to 45	0.090	*0.135*	0.180
	Over 45 to 90	0.135	0.180	0.225
	Over 90 to 135	0.180	0.225	0.270
	Over 135	0.225	0.270	0.315
Over 10 to 14, inc.	Up to 60	0.120	*0.180*	0.240
	Over 60 to 120	0.180	0.240	0.300
	Over 120 to 180	0.240	0.300	0.360
	Over 180	0.300	0.360	0.420
Over 14 to 18, inc.	Up to 80	0.160	*0.240*	0.320
	Over 80 to 160	0.240	0.320	0.400
	Over 160 to 240	0.320	0.400	0.480
	Over 240	0.400	0.480	0.560
Over 18 to 22, inc.	Up to 100	0.200	*0.300*	0.400
	Over 100 to 200	0.300	0.400	0.500
	Over 200 to 300	0.400	0.500	0.600
	Over 300	0.500	0.600	0.700
Over 22 to 28, inc.	Up to 125	0.250	*0.375*	0.500
	Over 125 to 250	0.375	0.500	0.625
	Over 250 to 375	0.500	0.625	0.750
	Over 375	0.625	0.750	0.875

1. These tolerances may be reduced 50 percent by individually grinding, measuring, testing and sorting the springs, but this procedure increases production time and cost.

2. These tolerances should be doubled for compression springs that do not have ground ends and for the usual run of commercial extension springs.

3. The values in italics are standard when the wire diameter or number of coils is not predetermined, or where there is doubt.

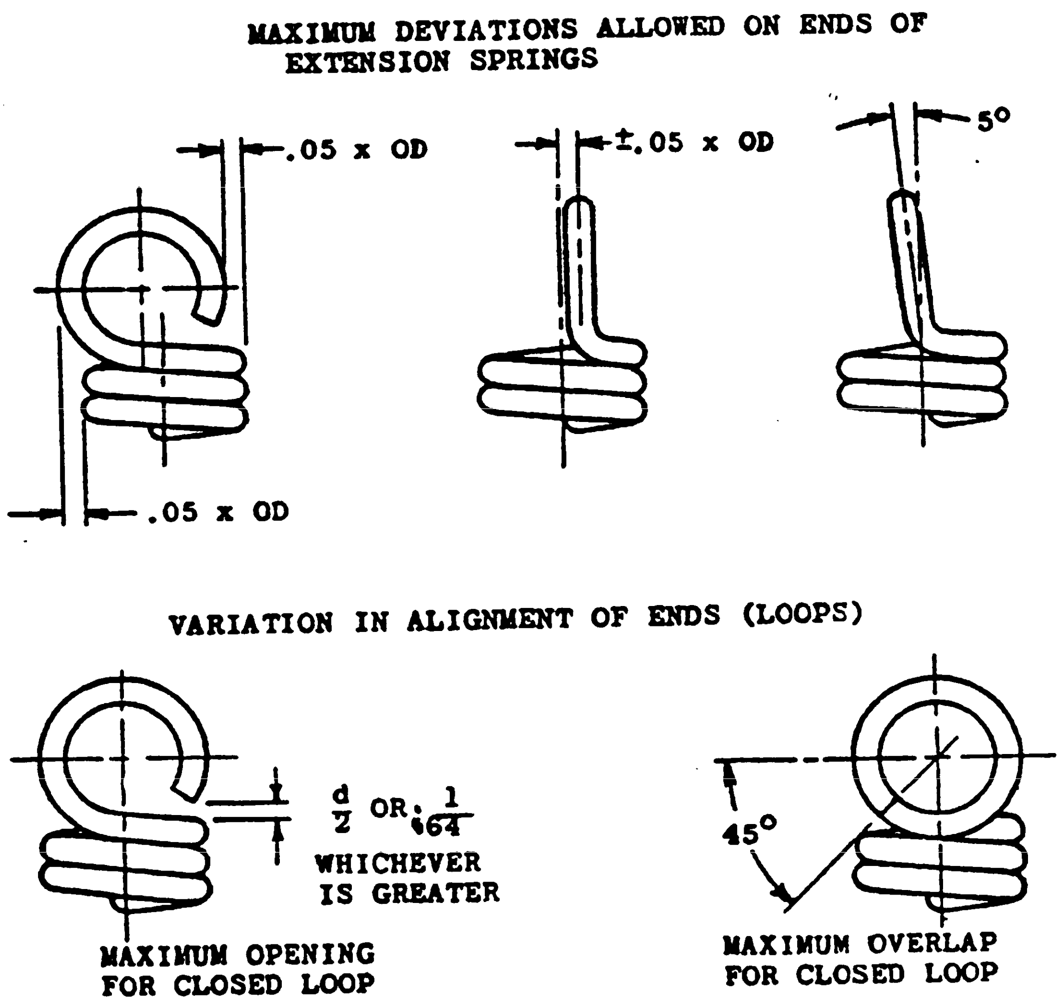

MAXIMUM DEVIATIONS ALLOWED ON ENDS OF EXTENSION SPRINGS
.05 x OD
±.05 x OD
5°
.05 x OD
VARIATION IN ALIGNMENT OF ENDS (LOOPS)
d/2 OR 1/64
WHICHEVER IS GREATER
MAXIMUM OPENING FOR CLOSED LOOP
45°
MAXIMUM OVERLAP FOR CLOSED LOOP

FIGURE 63

materials are not ordinarily subject to hydrogen embrittlement during electroplating.

37. TOLERANCES FOR SPRINGS. Tables that follow present tolerances which can be obtained under normal manufacturing methods. Springs having closer tolerances can be produced but at greatly increased cost. A more liberal tolerance should be specified whenever possible.

TABLE XX. *Tolerances on squareness of grinding on end coils for compression springs*

Number of coils per inch of length (Number of coils ÷ length)	Maximum tolerance from squareness to axis		
	Spring index (OD/d)		
	Up to 6 (small)	6 to 10 (average)	10 to 16 (large)
Up to 7	3°	4°	5°
Over 7 to 12	4°	5°	6°
Over 12 to 17	5°	6°	7°
Over 17	6°	7°	8°

These tolerances in degrees may be reduced 50 per cent by individually grinding, measuring, testing, and sorting the springs, but this increases production time and cost. Grinding end coils of springs made from wire under $\frac{1}{32}$ inch diameter is usually unnecessary and frequently is expensive.

37.1 TOLERANCE ON LOADS. The recommended tolerance on all loads is plus or minus 10 per cent and should be specified as shown on the drawing requirement charts. However, when the deflection from the free length to the first load is small, the load tolerance must be large and should be considered proportional to the tolerances that apply to the free length. For example, if the deflection is only $\frac{1}{8}$ inch and the tolerance on the free length is $\frac{1}{32}$ inch, then it follows that the tolerance on the first load should be plus or minus 25 per cent ($\frac{1}{32}$=25 per cent of $\frac{1}{8}$), otherwise the tolerance on the free length could not be taken.

Where a specific amount of initial tension is required for extension springs, it should be specified in pounds or ounces with a recommended tolerance of plus or minus 15 per cent, but it usually is specified as a REFERENCE.

Smaller load tolerances than these usually require testing and adjusting each spring and add considerably to the manufacturing time and to the cost.

TABLE XXI

Table of Third and Fourth Powers of Wire Diameters

d	d^3	d^4	d	d^3	d^4	d	d^3	d^4	d	d^3	d^4
.001	.000000001	.000000000001	.051	.00013265	.0000067652	.101	.00103030	.00010406	.151	.00344295	.00051989
.002	.000000008	.000000000016	.052	.00014061	.0000073116	.102	.00106121	.00010824	.152	.00351181	.00053379
.003	.000000027	.000000000081	.053	.00014888	.0000078905	.103	.00109273	.00011255	.153	.0035816	.00054798
.004	.000000064	.000000000256	.054	.00015746	.0000085031	.104	.00112486	.00011699	.154	.0036523	.00056245
.005	.000000125	.000000000625	.055	.00016638	.0000091506	.105	.00115762	.00012155	.155	.0037239	.00057720
.006	.000000216	.000000001296	.056	.00017562	.0000098345	.106	.00119102	.00012625	.156	.0037964	.00059224
.007	.000000343	.000000002401	.057	.00018519	.0000105560	.107	.00122504	.00013108	.157	.0038699	.00060757
.008	.000000512	.000000004096	.058	.00019511	.0000113165	.108	.00125971	.00013605	.158	.0039443	.00062320
.009	.000000729	.000000006561	.059	.00020538	.0000121174	.109	.00129503	.00014116	.159	.0040197	.00063913
.010	.000001000	.000000010000	.060	.00021600	.0000129600	.110	.00133100	.00014641	.160	.0040960	.00065536
.011	.000001331	.000000014641	.061	.00022698	.0000138458	.111	.0013676	.00015181	.161	.0041733	.00067190
.012	.000001728	.000000020736	.062	.00023833	.0000147763	.112	.0014049	.00015735	.162	.0042515	.00068875
.013	.000002197	.000000028561	.063	.00025005	.0000157530	.113	.0014429	.00016305	.163	.0043307	.00070591
.014	.000002744	.000000038416	.064	.00026214	.0000167772	.114	.0014815	.00016890	.164	.0044109	.00072339
.015	.000003375	.000000050625	.065	.00027463	.0000178506	.115	.0015209	.00017490	.165	.0044921	.00074120
.016	.000004096	.000000065536	.066	.00028750	.0000189747	.116	.0015609	.00018106	.166	.0045743	.00075933
.017	.000004913	.000000083521	.067	.00030076	.0000201511	.117	.0016016	.00018739	.167	.0046575	.00077780
.018	.000005832	.000000104976	.068	.00031443	.0000213814	.118	.0016430	.00019388	.168	.0047416	.00079659
.019	.000006859	.000000130321	.069	.00032851	.000022667	.119	.0016852	.00020053	.169	.0048268	.00081573
.020	.000008000	.000000160000	.070	.00034300	.0000240100	.120	.0017280	.00020736	.170	.0049130	.00083521
.021	.000009261	.000000194481	.071	.00035791	.0000254117	.121	.0017716	.00021436	.171	.0050002	.00085504
.022	.000010648	.000000234256	.072	.00037325	.0000268739	.122	.0018158	.00022153	.172	.0050884	.00087521
.023	.000012167	.00000027984	.073	.00038902	.0000283982	.123	.0018609	.00022889	.173	.0051777	.00089575
.024	.000013824	.00000033178	.074	.00040522	.0000299866	.124	.0019066	.00023642	.174	.0052680	.00091664
.025	.000015625	.00000039062	.075	.00042188	.0000316406	.125	.0019531	.00024414	.175	.0053594	.00093789
.026	.000017576	.00000045698	.076	.00043898	.0000333622	.126	.0020004	.00025205	.176	.0054518	.00095951
.027	.000019683	.00000053144	.077	.00045653	.0000351530	.127	.0020484	.00026014	.177	.0055452	.00098151
.028	.000021952	.00000061466	.078	.00047455	.0000370151	.128	.0020972	.00026844	.178	.0056398	.00100388
.029	.000024389	.00000070728	.079	.00049304	.0000389501	.129	.0021467	.00027692	.179	.0057353	.00102663
.030	.000027000	.00000081000	.080	.00051200	.0000409600	.130	.0021970	.00028561	.180	.0058320	.00104976
.031	.000029791	.00000092352	.081	.00053144	.0000430467	.131	.0022481	.00029450	.181	.0059297	.00107328
.032	.000032768	.00000104858	.082	.00055137	.0000452122	.132	.0023000	.00030360	.182	.0060286	.00109720
.033	.000035937	.00000118592	.083	.00057179	.0000474583	.133	.0023526	.00031290	.183	.0061285	.00112151
.034	.000039304	.00000133634	.084	.00059270	.0000497871	.134	.0024061	.00032242	.184	.0062295	.00114623
.035	.000042875	.00000150062	.085	.00061412	.0000522006	.135	.0024604	.00033215	.185	.0063316	.00117135
.036	.000046656	.00000167962	.086	.00063606	.0000547008	.136	.0025155	.00034210	.186	.0064349	.00119688
.037	.000050653	.00000187416	.087	.00065850	.0000572898	.137	.0025714	.00035228	.187	.0065392	.00122283
.038	.000054872	.00000208514	.088	.00068147	.0000599695	.138	.0026281	.00036267	.188	.0066447	.00124920
.039	.000059319	.00000231344	.089	.00070497	.0000627422	.139	.0026856	.00037330	.189	.0067513	.00127599
.040	.000064000	.00000256000	.090	.00072900	.0000656100	.140	.0027440	.00038416	.190	.0068590	.0013032
.041	.000068921	.00000282576	.091	.00075357	.0000685750	.141	.0028032	.00039525	.191	.0069679	.0013309
.042	.000074088	.00000311170	.092	.00077869	.0000716393	.142	.0028633	.00040659	.192	.0070779	.0013590
.043	.000079507	.00000341880	.093	.00080436	.0000748052	.143	.0029242	.00041816	.193	.0071891	.0013875
.044	.000085184	.00000374810	.094	.00083058	.0000780749	.144	.0029860	.00042998	.194	.0073014	.0014165
.045	.000091125	.00000410062	.095	.00085738	.0000814506	.145	.0030486	.00044205	.195	.0074149	.0014459
.046	.000097336	.00000447746	.096	.00088474	.0000849347	.146	.0031121	.00045437	.196	.0075295	.0014758
.047	.000103823	.00000487968	.097	.00091267	.0000885293	.147	.0031765	.00046695	.197	.0076454	.0015061
.048	.000110592	.00000530842	.098	.00094119	.0000922368	.148	.0032418	.00047979	.198	.0077624	.0015370
.049	.000117649	.00000576480	.099	.00097030	.0000960596	.149	.0033079	.00049288	.199	.0078806	.0015682
.050	.000125000	.00000625000	.100	.00100000	.0000001	.150	.0033750	.00050625	.200	.0080000	.0016000

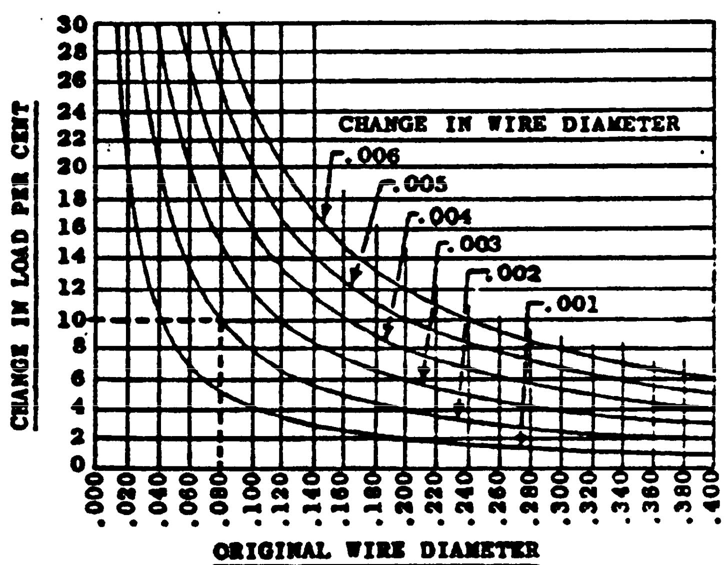
TABLE XXII. *Load changes due to slight wire changes*

INTRODUCTION: It is frequently necessary to make a slight change to the wire diameter of a spring, either to use a wire size available or to change the load to meet a specified requirement. Adding only .001 in. to .040 in. wire (thereby using .041 in. instead of .040 in.) increases the load 10 per cent. Adding .001 in. to $^3/_{16}$ in. wire, however, changes the load only 2 per cent, but a .005 in. addition increases the load 10 per cent. This data is for compression and extension springs.

CURVES: The curves below show the per cent change in loads due to slight changes in wire diameters. The curves are suitable for compression, extension and torsion springs using any material, with wire diameters between .015 in. and .375 in. inclusive.

EXAMPLE: If a spring that is supposed to be made from .080 in. wire, is actually made from .082 in. wire, the .002 in. *heavier* wire causes a *heavier* load. The per cent increase in load is 10 per cent as shown by the intersection of the vertical line .080 in. and the curve .002 in. (as shown dotted).

Similarly if .078 in. wire is used, the .002 in *lighter* wire causes a *lighter* load of 10 per cent (same curve). If the original load must be maintained, it would be necessary to add 10 per cent more coils if the heavier wire is used or 10 per cent less coils with the lighter wire.



Note: Changing the wire size also changes the solid height, body length, stress, etc.

TABLE XXIII. *Causes of spring failure. Listed in sequence according to frequency of failure.*

Spring failure may be cause by breakage, high permanent set or loss of load. Group I lists the causes that occur most frequently. Group II contains the less frequent causes and Group III lists causes that occur occasionally.

Group	Cause	Comment
GROUP I	HIGH STRESS	The majority of spring failures are due to high stresses caused by large deflections and high loads. Highstresses should be used only for statically loaded springs. Low stresses lengthen fatigue life.
	HYDROGEN EMBRITTLE-MENT	Improper electro-plating methods and acid cleaning of springs, without proper baking treatment, cause spring steels to become brittle, and is a frequent cause of failure. Non-ferrous springs are immune.
	SHARP BENDS & HOLES	Sharp bends on extension, torsion and flat springs and holes or notches in flat springs, cause high concentration of stress resulting in failure. Bend radii should be as large as possible, and tool marks avoided.
	FATIGUE	Repeated deflections of springs, especially above 1,000,000 operations, even with medium stresses, may cause failure. Low stresses should be used for severe operating conditions.
GROUP II	SHOCK LOADING	Impact, shock, and rapid loading cause far higher stresses than those computed by the regular spring formulas. High carbon spring steels do not withstand shock loading as well as alloy steels.
	CORROSION	Slight rusting or pitting caused by acids, alkali, galvanic corrosion, stress corrosion cracking, or corrosive atmosphere weakens the material and causes higher stresses in the corroded area.
	FAULTY HEAT TREATMENT	Keeping spring materials at the hardening temperature for longer periods than necessary causes an undesirable growth in grain structure resulting in brittleness even though the hardness may be correct.
	FAULTY MATERIAL	Poor material containing inclusions, seams, slivers, and flat material with rough, slit, or torn edges cause early failure. Overdrawn wire, improper hardness, and poor grain structure also result in early failure.
GROUP III	HIGH TEMPERA-TURE	High temperatures reduce spring temper (or hardness), lowers the modulus of elasticity thereby causing lower loads, reduces the elastic limit and increases corrosion. Corrosion resisting or nickel alloys should be used.
	LOW TEMPERA-TURE	Temperatures below — 40° F lessen the ability of carbon steels to withstand shock loads. Carbon steels become brittle at — 70° F. Corrosion resisting, nickel or non-ferrous alloys should be used.
	FRICTION	Close fits on rods or in holes result in a wearing away of material and occasional failure. The outside diameters of compression springs expand during deflection but they become smaller on torsion springs.
	OTHER CAUSES	Enlarged hooks on extension springs increases the stress at the bends. Carrying too much electrical current will cause failure. Welding and soldering frequently destroy the spring temper. Tool marks, nicks, and cuts often become stress raisers. Deflecting torsion springs outwardly causes high stresses. Winding them tightly causes binding on supporting rods. High speed of deflection, vibration and surging due to operation near natural periods of vibration or their harmonics, cause increased stresses.

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